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U. S. NAVY BATTLE DAMAGE PICTURES FROM WORLD WAR II.

BY COMMANDER ERNEST C. HOLTZWORTH, U. S. NAVY.

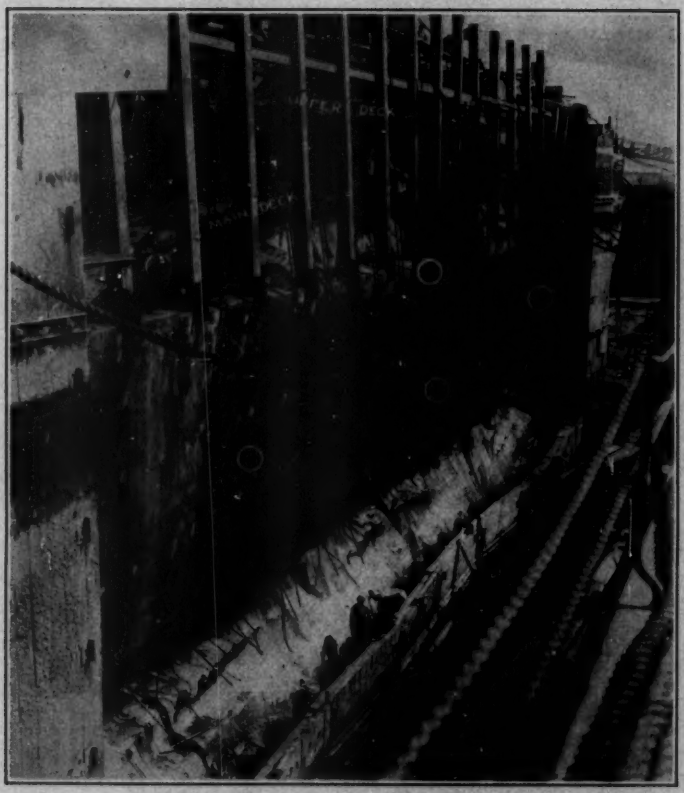
PART 2.

The following collection of photographs, the second of a series, is devoted entirely to ships damaged at Pearl Harbor on 7 December 1941. All are official U. S. Navy photographs. A brief account of the circumstances involved in the damage each ship incurred is included as a matter of general interest.

Photo 1: The U.S.S. *West Virginia* (BB48) in drydock at Pearl Harbor, 11 June 1942, looking aft from frame 64. She was struck by a total of seven aircraft torpedoes on the port side and two bombs. Of the torpedoes, one struck the rudder and four struck the armor belt, bilging the port torpedo defense voids and causing a heavy port list. While listed about 22 degrees two additional torpedoes hit the side, one well above the armor belt and the other at the top of the armor belt, demolishing the shell over a wide area. Both bombs (converted heavy AP projectiles) were duds, one penetrating the roof of No. 3 turret and breaking up in the gun room, and the other coming to rest intact on the second deck amidships. In the photo the points of impact of five torpedoes are marked by white circles. Note the crushing of the outer torpedo defense voids below the armor belt and the outboard edges of the collapsed main and upper (forcastle) decks. Maximum distortion of the armor belt (13½ inches thick at the top) was about 10 inches outboard at the top and about 33 inches inboard at the bottom. The torpedo defense system was not penetrated. The ship was sunk by destruction of the port shell above the armor combined with the heavy list which permitted extensive flooding on second and third decks. She did not capsize, as did *Oklahoma*, because of prompt flooding of starboard torpedo defense voids which returned her almost to the upright as she settled.

Some idea of the magnitude of the salvage job is obtained from the photo. It was necessary to place a composite patch from the bilge to within four feet of the upper deck level, between frames 61 and 96 and another forward from frames 43 to 52, a total length of about 196 feet. The patches were placed in sections while the ship rested on the bottom with the waterline some five feet below the upper deck. The vertical channels above the armor belt, a part of the backing structure, and the end sections of the after patch may be seen in the photo.

West Virginia was floated on 17 May 1942 and placed in drydock on 9 June 1942 when the photo was taken.



U. S. S. "WEST VIRGINIA" (BB48).

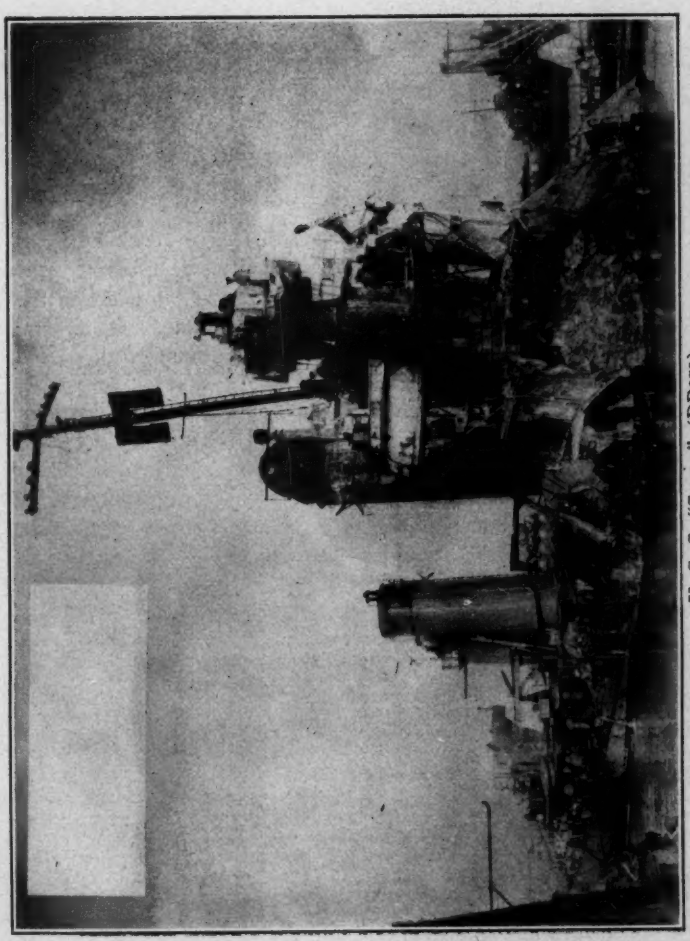
Photo 2: The U.S.S. *Shaw* (DD373) in floating drydock (largely submerged) at Pearl Harbor on 9 December 1941. At the time of the attack *Shaw* was in YFD2 receiving repairs. A flight of dive bombers scored three hits on *Shaw*.

One of these penetrated *Shaw's* forward superstructure and passed clear of the ship, detonating on the floor of the dock abreast the forward fuel oil tanks on the starboard side. Great quantities of oil flooded the dock. The other two bombs detonated within the ship at about the main deck level in the vicinity of the wardroom. A raging fire soon engulfed the forward portion of the ship back to No. 2 fireroom. Despite efforts to submerge the dock (which were not immediately successful due to damage to the dock) *Shaw's* forward magazines exploded, completely severing the bow at about frame 65, just forward of the bridge.

In the photo the mass of debris on the forecastle level just forward of the STS director tower should be noted. A large portion is slag from molten aluminum. The STS truck leading upward to the base of the director stood up very well and prevented general collapse of the superstructure.

The forward fireroom flooded completely through fragment holes in the forward bulkhead. The after fireroom flooded only to a depth of one or two feet as a result of minor leaks. The flooded fireroom put the *Shaw* well down forward as shown. When this space was dewatered she floated with almost no trim.

She was placed on the marine railway stern first where the stub bow, which received so much publicity in the early months of the war, was fitted. She returned to Mare Island under her own power where a new bow structure was attached. Oddly enough, despite the violence of the magazine explosion, she suffered almost no damage to machinery. Shafts remained in satisfactory alignment for the trip back although the bottom shell was wrinkled aft of the after reduction gear foundations.



U. S. S. "SHAW" (DD373).

Photo 3: The U.S.S. *Helena* (CL50) in drydock at Pearl Harbor on 14 December 1941. Berthed portside to a pier, she was struck by a deep-running Japanese aircraft torpedo on the starboard side at frame 74 in way of the forward engineroom. The *Oglala* (CM4) was berthed alongside and the torpedo passed underneath her to strike *Helena.*

The photo shows in some detail what happens to an engineroom when struck by a torpedo. Note that No. 1 shaft is severed completely at the coupling aft of the reduction gear. The floor plates have been driven into the teeth of the reduction gear, with only remnants of the reduction gear casing visible. The major portion of the casing was blown upward and forward, demolishing the forward main distribution switchboard. Almost all of the machinery for No. 1 main unit (the starboard outboard plant) required replacement. The port main unit was undamaged except by immersion. Of the ships seriously damaged at Pearl Harbor, *Helena* was the first to be placed in drydock.

U. S. S. "HELENA" (CL50).

Photo 4: The U.S.S. *Cassin* (DD372) and the U.S.S. *Downes* (DD375) in drydock No. 1 at Pearl Harbor on 7 December 1941 with the U.S.S. *Pennsylvania* (BB38) in the background. All three ships were in drydock at the time.

The two unfortunate destroyers had a greater variety of mishaps befall them in the short period of 30 minutes than has occurred to any other destroyers in the whole course of the war. At the time, both were undergoing extensive overhauls, including among other jobs the replacement of shell plates in the bow and opening other plates in way of the machinery spaces. Plates in both locations on both ships were off at the time. *Cassin* was hit by three bombs, two of which caused extensive damage to *Downes*. In addition, *Downes* was hit by a fourth bomb. Oil tanks were opened and free oil in the dock was ignited. Both ships were gutted by fire, in the course of which ready service ammunition exploded, a magazine fire occurred, torpedo warheads burned and a heavy explosion in the vicinity of the starboard torpedo mount on *Downes* further seriously damaged that ship. When the dock was flooded to protect *Pennsylvania* and assist in fire fighting, both ships lifted initially, then as they quickly flooded they settled improperly on the blocks. *Cassin* (to the right in the photo) capsizing to starboard against *Downes*. The hulls of both vessels were complete losses, although they were floated after extensive patching to permit salvage of machinery.

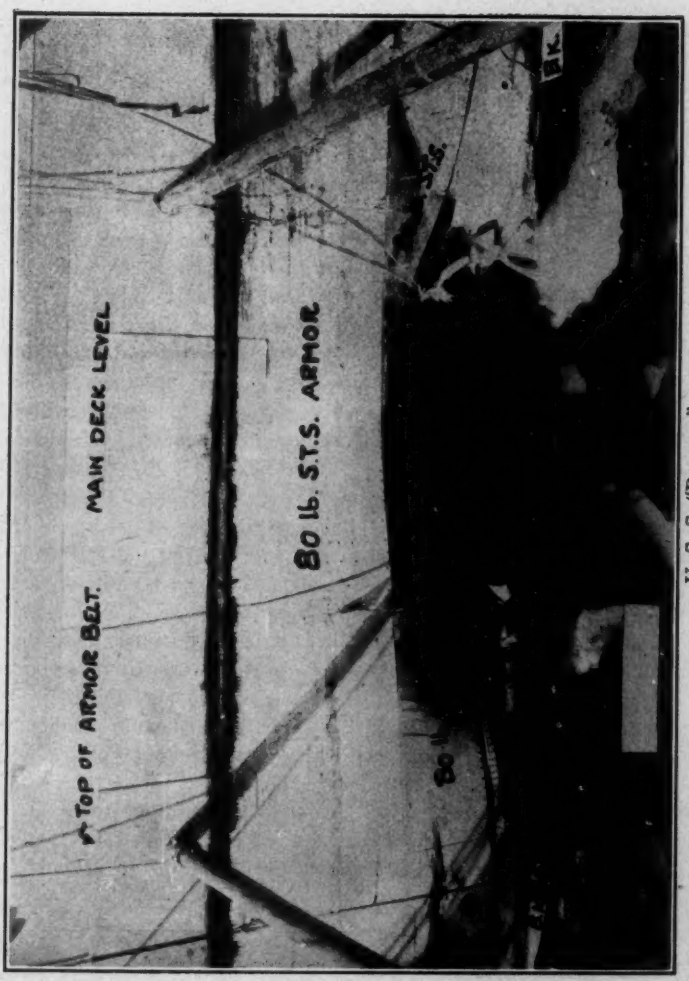
Remarkably, almost all machinery was salvaged and reinstalled in hulls of the same name and number built at Mare Island. The total value of machinery salvaged was much greater than that of the hulls. *Pennsylvania* received minor damage only from a single topside bomb hit.

U. S. S. "CASSIN" (DD372). U. S. S. "DOWNES" (DD375).
U. S. S. "PENNSYLVANIA" (BB38).

Photo 5: The U.S.S. *Raleigh* in drydock at Pearl Harbor on 3 January 1942. She was struck by a Japanese aircraft torpedo on the port side in way of No. 2 fireroom, flooding No. 1 fireroom and the forward engineroom as well. Approximately an hour later an armor piercing bomb (converted heavy caliber projectile) struck the after deck house and passed completely through the vessel, detonating at a considerable distance from the hull. Several after spaces on three levels flooded as a result. The *Raleigh* was an old ship with watertight integrity considerably below present standards and progressive flooding threatened her survival for several days. Only the most strenuous efforts by her crew kept her afloat. She was moved to the Navy Yard the latter part of December and placed in drydock on 3 January, when the photo was taken.

The hole in the shell is somewhat smaller than on other ships damaged by torpedoes at Pearl Harbor. It was about 24 feet in length and about 20 feet along the girth. *Raleigh* had, in effect, a double hull in way of the impact with 80-pound (2 inch) STS over a 40-pound (1 inch) STS or HTS shell, both courses being riveted together. These heavy plates were forced inward, forming a deep indentation and shearing rivets in seams and butts and tearing a few plates.

Raleigh received temporary repairs at Pearl Harbor and then returned to Mare Island where she was repaired permanently.

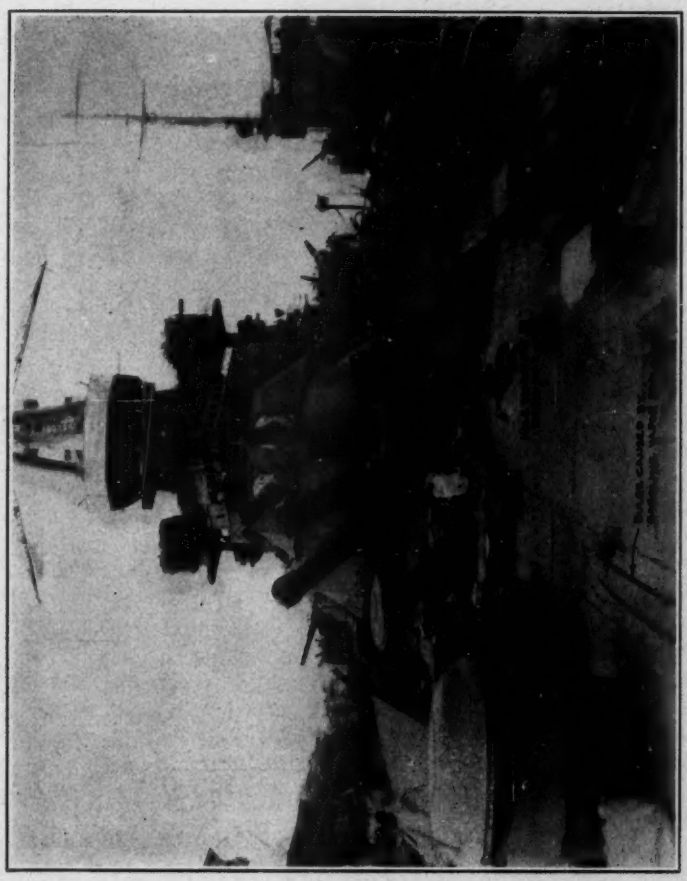


U. S. S. "RALEIGH".

Photo 6: Bomb damage to the forecastle deck of U.S.S. *Nevada* (BB36), photo taken on 12 December 1941. She was struck initially by one torpedo on the port side between Nos. 1 and 2 turrets, which put her somewhat down by the bow, but otherwise having little effect. In passing down the harbor she suffered a severe bombing attack, being struck by five bombs. Three of these hit the forecastle and two hit amidships.

Of the three bombs which hit forward, the impact holes of two are shown in the photo. The one shown as bomb No. 3 detonated just above the second deck, blowing out the watertight bulkhead at frame 30 and damaging other watertight structure in the vicinity. All bulges in the deck were caused by this bomb. It also started a bad fire in interior spaces. Bomb No. 2 passed down to the second platform and detonated at the port shell at frame 8, causing considerable flooding forward and opening the gasoline tank, later resulting in a gasoline vapor explosion. The other bomb forward struck at frame 15 and passed out the starboard side at the second deck, detonating close aboard below the waterline at frame 16 and flooding many other spaces. Flooding of forward magazines because of the fire, plus flooding from the torpedo and the two bomb hits, put the waterline above the second deck at the bow. Water ran aft on the second deck and poured down to lower spaces via a central ventilation plenum chamber. *Nevada* was beached but, despite the most strenuous efforts of her crew, gradually flooded almost completely.

When the bomb holes were patched she was dewatered and placed in drydock at Pearl Harbor. She proceeded to Puget Sound for permanent repairs and modernization.

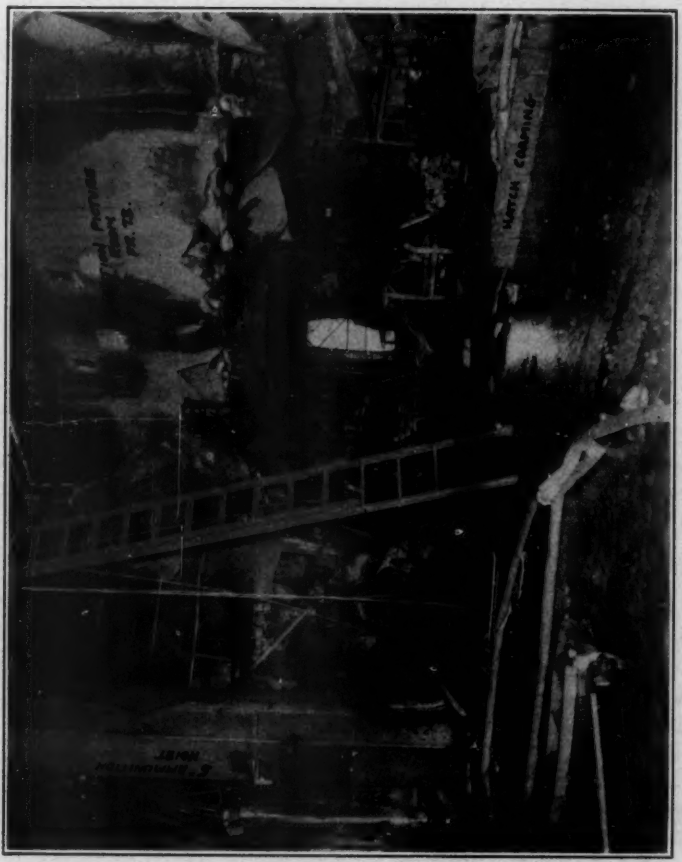


U. S. S. "NEVADA" (BB36).

Photo 7: Bomb damage in the forward end of the hanger of the seaplane tender *Curtiss* (AV4), photo taken 9 December 1941. A semi-armor piercing bomb struck the boat deck to starboard of No. 2 smokepipe and penetrated three decks, detonating high order in contact with the main deck just to port of the hatch shown in the photo. One plane parked at the after end of the hangar undergoing repairs was ignited and contributed to a severe fire.

In the photo, several shop enclosures under the motion picture room were completely demolished. Note the fragment damage to the ammunition hoist. The main deck plating is of 20-pound (½ inch) medium steel. Fragments passed down through the second and third decks into the after engineroom severing steam lines and other piping.

Curtiss has the unique distinction of being the target for the first Japanese plane crash, probably a suicider. During the final stages of the attack on Pearl Harbor a plane crashed against the forward starboard crane, above the boat deck, adjacent to the bomb hit described above. Gasoline ran down through the holes in the various upper decks greatly augmenting the fire in the hangar. The crane was rendered inoperable. *Curtiss* performed her own temporary repairs in a period of a few days and joined the operating forces. She was not permanently repaired until May, 1942 at Pearl Harbor.



"CURTISS" (AV4).

SURFACE FINISH OF REDUCTION GEAR TEETH.

By J. A. Davies.*

The experience gained during the past few years, when propulsion reduction gears for both naval and merchant ships were operated under conditions more severe than is usual in peacetimes, has not only concentrated attention on the load carrying capacity of gears as at present designed but has also emphasized the opinion, now widely accepted, that the smoother the surface of the gear teeth the better able are they to carry the rated loads without encountering difficulties from excessive pitting.

A need of long standing in the engineering profession has been a suitable means for expressing surface finishes of machined parts. Each manufacturer has hitherto developed a system of finish marks peculiar to his own shops. It is doubtful if any two such systems are alike. The need of a standard has been recognized, and a committee of the American Standards Association is now giving consideration to the matter.

A determination of the smoothness of a machined surface can be made by the use of a profilometer and the quality of finish, or "surface roughness" as it is termed in the literature, expressed in micro-inches. However, except to an expert operator, this is not an easy measurement to make on a gear tooth, particularly where the pitch of the teeth is small, but it is possible to obtain a satisfactory visual inspection of the surface of gear teeth by the use of film impressions, known in the trade as "Faxfilms". Those films can be mounted and used as a slide. When inserted in a projecting machine, enlargements up to forty to one can be thrown on to a screen or on a white or cream painted office wall, and the condition and relative smoothness of the tooth surface can be examined conveniently and compared with other gears, or with a film of a so-called "ideal" finish.

The usual procedure adopted to produce reduction gears is to cut the teeth in a hobbing machine or a shaping machine similar to the "Sykes" type gear cutter, and then "lap" the mating sur-

* Manager, Marine Turbine Engineering Dept. Westinghouse Electric Corporation, Lester Branch P. O., Philadelphia 13, Pa.

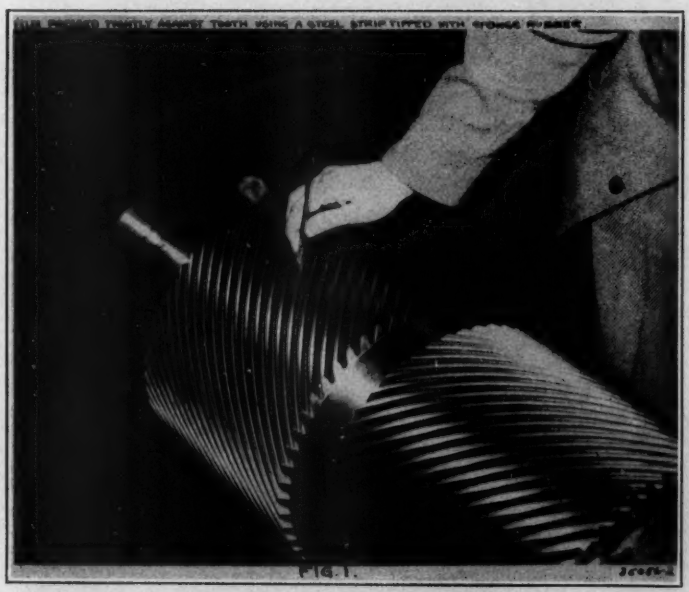


FIG. 1.

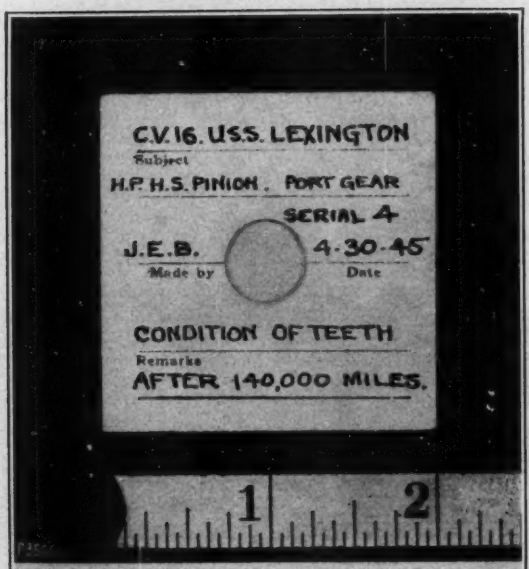

C.V.16. U.S.S. LEXINGTON
Subject
H.P. H.S. PINION. PORT GEAR
SERIAL 4
J.E.B.
Made by
4-30-45
Date
CONDITION OF TEETH
Remarks
AFTER 140,000 MILES.
1
2

FIG. 2.

FIG. 3

FIG. 4.
HOBBED- NO OTHER FINISH.
14 X FULL SIZE.

FIG. 5.
HOBBED TOOTH, FINISHED BY LAPPING, USING GLASS POWDER AND OIL.
THIS GEAR HAS NOT BEEN IN SERVICE.
14 X FULL SIZE.

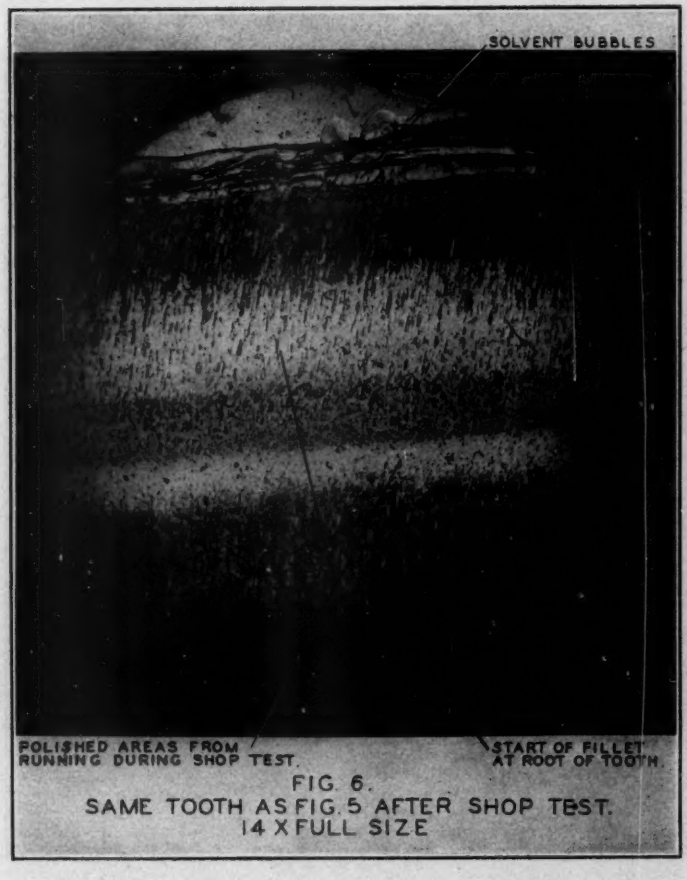


FIG. 6.
SAME TOOTH AS FIG. 5 AFTER SHOP TEST.
14 X FULL SIZE

SOLVENT BUBBLES ALONG EDGE OF TOOTH
WIDTH OF CHAMFER AT TOP OF TOOTH
SCRATCHES FROM DIRT IN THE OIL
HOB MARKS IN FILLET AT ROOT OF TOOTH
DIRT ON FILM

FIG 7
HOBBED – POLISHED BY LONG SERVICE.
DAMAGED BY DIRT IN THE OIL.
14 X FULL SIZE

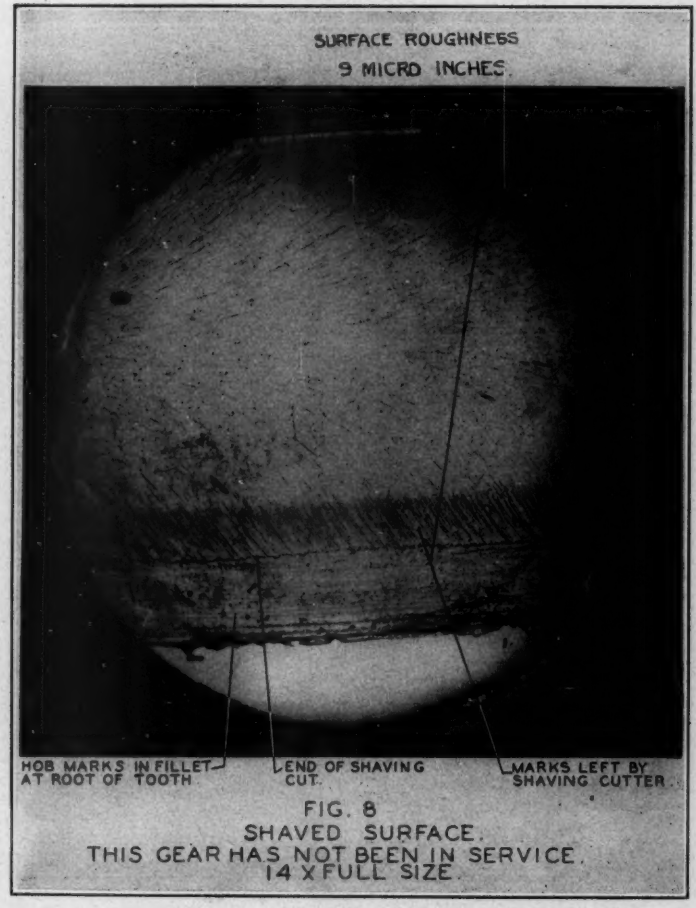


FIG. 8
SHAVED SURFACE.
THIS GEAR HAS NOT BEEN IN SERVICE.
14 X FULL SIZE.

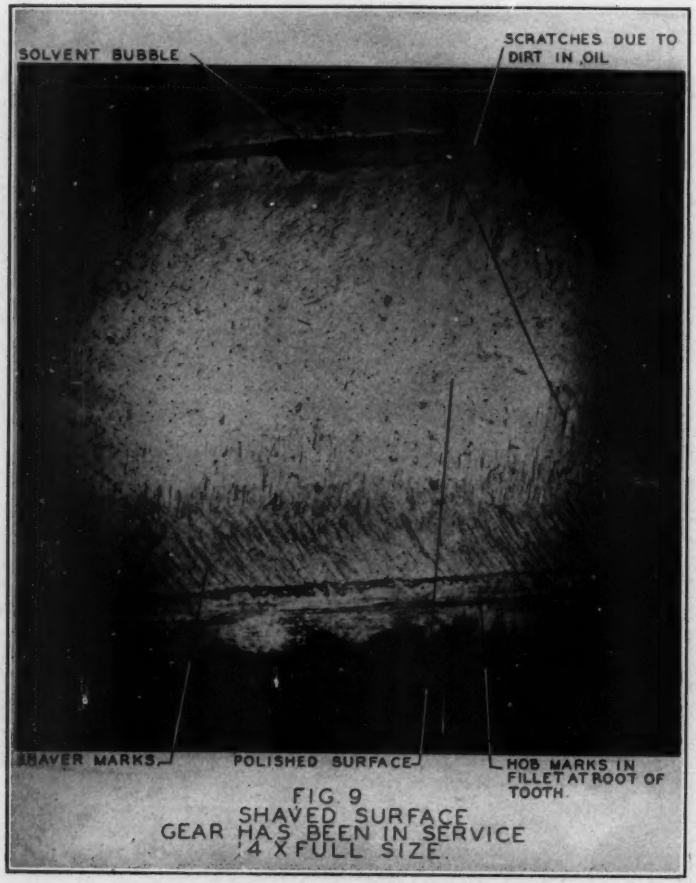


FIG. 9
SHAVED SURFACE
GEAR HAS BEEN IN SERVICE
4 X FULL SIZE.

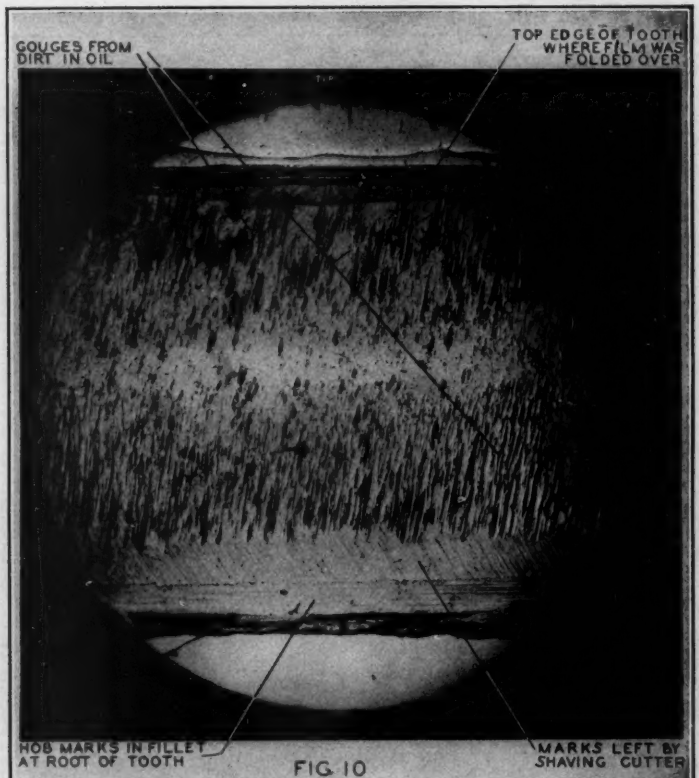

GOUGES FROM DIRT IN OIL
TOP EDGE OF TOOTH WHERE FILM WAS FOLDED OVER
HOB MARKS IN FILLET AT ROOT OF TOOTH
MARKS LEFT BY SHAVING CUTTER

FIG. 10

TEETH WERE SURFACE FINISHED BY SHAVING. DAMAGED ON TEST BY METAL PARTICLES IN THE OIL WHICH CAME THROUGH THE OIL SPRAYS. 14X FULL SIZE.

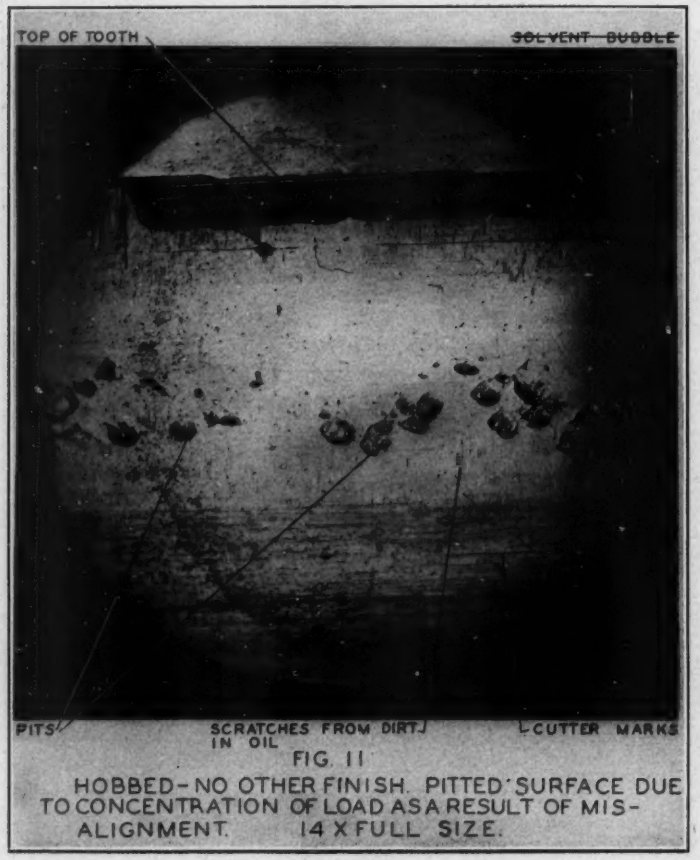


FIG. 11

HOBBED - NO OTHER FINISH. PITTED SURFACE DUE TO CONCENTRATION OF LOAD AS A RESULT OF MIS-ALIGNMENT. 14 X FULL SIZE.

FIG. 12
GALLED TOOTH SURFACE
14 X FULL SIZE

faces together, using an abrasive in oil, until the bearing surface on both the pinion and gear teeth extends uniformly from end to end of each helix.

A second method is to cut the teeth and then finish in a "shaving" machine. This shaving operation removes any slight irregularities introduced in the hobbing or shaping process and leaves the flanks of the teeth in a very smooth condition.

A third method is, of course, to hob or shape the teeth of the gear rotating parts and use them without any further surface finishing.

A variation of these schemes has been to burnish the teeth after finish machining by a process which is tantamount to a surface hardening operation.

Regardless of the method used, the "Faxfilms" will show to the experienced observer a true comparison of the smoothness of finish on teeth produced in these various ways. This method for recording the condition of the tooth surface is also of value for keeping a history of the gear to show changes that occur in service. If such a subsequent record is desired the teeth from which the original films were taken should, of course, be clearly marked so that the later films will cover the same part of the tooth surface.

The teeth to be examined must be clean and free from oil. A portion of the tooth surface is then coated with a solvent to which is immediately applied a strip of the recording film. (See Figure 1) It is to be noted that the film impression can be taken in full daylight. This film is then pressed tightly against the tooth and held in that position for from fifteen to thirty seconds. During this period the film faithfully registers every nook and cranny existing in the tooth surface at the spot that is being examined. Care must be exercised to squeeze out any bubbles and excess solvent between the film and the tooth surface as otherwise these bubbles will give the impression of defects in the picture. To locate the film before removing it from the tooth the end of the strip can be folded over the top edge of the tooth. This leaves a distinct line on the film making easy the identification as to position.

The film is then peeled off the tooth and carries with it the true impression picked up by the solvent which has adhered to the film. This film strip is mounted in a cardboard slide (see Figure 2) on which a record of the gear in question can be posted.

The slide is then ready for use in the special projecting machine made for the purpose, a view of which is shown in Figure 3.

Figure 4 is a photograph of a film magnified fourteen times and showing the surface of a gear tooth finished on the hobbing machine. In examining such films after they have been so considerably enlarged—projection enlargements as much as forty to one can be had—the observer is apt to form a pessimistic opinion of the tooth surface thus exhibited until he calls to mind the extent of the magnification. For instance, the tooth surface of Figure 4 has a profilometer reading of 24 to 40 micro-inches. The tooth surface is just as it came from the hobbing machine and has not been lapped or shaved, or had any of the other finishes that would improve the surface smoothness.

Figure 5 is an impression taken from a tooth which had first been finished on the hobbing machine and had then had the surface "lapped" by grinding together the pinion and gear using glass powder and oil. It might well be questioned whether the result has given any better load carrying surface than existed on the original hobbed tooth face. The scratches and minute depressions put into the tooth flank by the particles of abrasive material are clearly shown in the film, and here again the observer must bear in mind the size of the magnification to avoid erroneous conclusions. The profilometer reading on this tooth is 45 micro-inches.

Figure 6 is a film of the same tooth as shown in Figure 5, but made after the gear had been subjected to a shop running test. It will be evident from a study of this picture that the root and tip of the tooth flank have been subjected to more grinding than the surface in the region of the pitch line. This is reasonable in view of the fact that the teeth are of the involute contour and, of course, follow the usual pattern in meshing, there being the maximum sliding at the tip and the root, with merely rolling at the pitch line. The polished areas on the teeth which carry the load are clearly seen in the photograph.

Figure 7 shows a hobbed tooth that has seen lengthy service. The white areas on the tooth flank are well polished surfaces. The mark from the hob in the fillet at the root of the tooth can be plainly seen and the depth of engagement is also very clear.

It will be observed that the dark band at the top of the film is a 1/32 inch chamfer put on the tips of the teeth during manufacture. This film shows a number of scratches or slight gouges that

result from metal particles which are carried over with the oil and pass into the teeth from the oil sprays.

Figure 8 was made from a tooth that had been "shaved" after it had been finished in the hobbing machine. The gear had not yet been operated. While this surface is very smooth to the touch and has a surface roughness of only 9 micro-inches, the shaver marks can be clearly seen in the picture. The shaving operation removes metal only to the depth of engagement, terminating near the finish of the circular fillet at the root of the tooth. This line of demarcation is clearly seen in the film. Again it must be mentioned that the fourteen to one magnification accentuates slight marks that are not always visible by a direct inspection of the teeth.

Figure 9 is a film from a tooth surface of a large Navy gear after considerable service. This tooth was finished by "shaving", and the cutter marks can still be seen, particularly at the root of the tooth. The white surface, from near the tip of the tooth to well below the pitch line, is part of a highly polished bearing surface. A few marks from dirty oil are indicated in this film, particularly at the right hand side of the picture.

Figure 10 illustrates a gear which had been finished in the shop by the "shaving" process but was considerably damaged by dirt in the oil during a dock trial. The film was taken at the portion of the tooth which had received the heaviest scratches. A first impression from observing the teeth and before the "Faxfilm" was examined led to the thought that the teeth had galled, but on viewing the film it was clearly evident that galling had not taken place, the gouges in the tooth flank being the familiar ones caused by small metal particles, sand or similar hard abrasives coming over with the oil. It will be noted from this film that the heavier scratching is at the upper and lower portion of the face where sliding takes place. The marks in the region of the pitch line are much fewer in number and many have the appearance of straight indentations without dragging, which is natural in view of the action of the mating teeth.

Figure 11 shows the surface of a tooth which was originally hobbed, had no other surface finish before being put into operation, and eventually pitted as a result of a heavy concentration of load due to misalignment. This film also has a magnification of fourteen to one and, as will be seen, the pitting occurs in the proximity of the pitch line. The hobber marks can be seen in

the fillet at the root of the tooth and the whitish surface of the film represents the high polish on the working face of the tooth. The dark line across the top of the film is the slight chamfer on the edge of the teeth, mentioned in connection with a previous film. To the upper left in this chamfered area can be seen a spot where, evidently, a piece of metal ejected from a pit slid between the teeth and caused material to flow by the chamfer to the tip of the tooth.

Figure 12 depicts the surface of a tooth which had seen considerable service and due, it is thought, to failure of the oil supply, galling had occurred at the points of the pinion teeth and, of course, at the roots of the gear teeth. This type of trouble is by no means as frequent as it was some years ago, but the film impression gives a vivid picture of the damage which has occurred.

It will be obvious that the simple equipment required for producing films of the type described makes possible the presentation of clear and convincing reports from distant outposts to a headquarters office. While this article has been devoted to the use of films for inspection of gear tooth surfaces, the process can be and has been used on any surface the condition of which is subject to question.

A COMPARISON OF AUTOMATIC BOILER CONTROL SYSTEMS FOR MARINE POWER PLANTS.

By P. S. Dickey*

The United States Navy recently brought two German Destroyers of the latest design to this country for inspection and study by the U. S. Navy and by Marine and Stationary Power Engineers. One of the features which has aroused considerable interest is the automatic control system used in the fire rooms of these ships. This automatic feature has been given considerable attention in the press releases covering these ships.

Less well known is the fact that the French Battleship *Richelieu* which was in this country during 1943 for repair and overhaul likewise was equipped with automatic boiler control.

While more than 1200 ships of various types built in this country during the past six years have been equipped with automatic boiler control, the U. S. Navy has used boiler control on only a few ships of special classification or on those ships built by the U. S. Maritime Commission or by private operators and released to and operated by the Navy.

In view of the above, a comparison of the automatic controlled fire rooms used by the German and French Navies with similar equipment used on American Vessels should be of interest. Likewise, there should be some justification for the fact that the U. S. Navy has not generally adopted the automatic boiler equipment used in the more modern vessels of the German and French Navies.

Fire Room Machinery—German Destroyers.

The two German destroyers, one large and one small are provided with geared turbine drive on two shafts with total propulsion horsepower in the order of 70,000 and 35,000 Hp. respectively. The larger unit is equipped with six boilers in three fire rooms and the smaller unit with four boilers in two fire rooms.

* Chief Engineer, Bailey Meter Company.

The boilers are of the natural circulation type and are suitable for generation of about 100,000 pounds of steam per hour each at steam conditions of 1000 psi and 750 degree F. Each pair of boilers in a fire room is served by two turbine driven forced draft fans each discharging air under pressure through an air preheater to the burner windbox. Dampers between the preheater outlet and the burner box regulate the flow of air for combustion.

The boilers on the larger destroyer are each equipped with one Saacke Wide Range-Rotary Cup type Oil Burner at each end of the boiler furnace. A constant speed turbine driven air compressor furnishes about 1200 cfm of primary air at a pressure of about 35 inch H_2O above fireroom pressure to rotate the burner cup and to assist in atomizing and mixing the oil with combustion air. An oil pump geared to the compressor furnishes fuel oil at a pressure of about 120 pounds. These burners are reputed to have an operating range of 20 to 1.

Oil Burners—Small Destroyer.

Each boiler of the smaller destroyer is equipped with five mechanical atomizing burners of conventional design on either end of the furnace making a total of ten. In order to achieve the necessary wide range of operation, means is provided for automatically changing the number of burners in service. Two burners, one in the center of the burner box at each end of the furnace, are left in service at all times. A small hydraulic cylinder is used on each of the remaining burners with linkage between this cylinder and the fuel oil shut-off valve and secondary air register-type damper. Control is provided to start and stop the burner at predetermined fuel oil pressures. Ignition of these burners which are automatically put in service is obtained from the center burners which operate continuously.

An added novel feature is the use of a chemical torch for the two center burners. This torch consists of a small cartridge which is electrically ignited through an interlocking switch connected to the fuel oil line to provide flame for a short time when igniting the center burners.

Automatic Control.

The automatic control system shown in Figure 1 is hydraulically operated with mechanisms similar to the Askania System

used in this country. The control consists of a steam pressure responsive device which actuates an hydraulic cylinder to position a cam shaft in accordance with steam pressure changes. This cam shaft positions a regulating valve in the fuel oil line to each burner. A separate regulator maintains a constant fuel oil pressure ahead of the fuel regulating valves so that fuel flow to the burners is proportional to the regulating valve position. A handwheel is provided on the front of the control panel to actuate the cam shaft for manual operation or in case the control is inoperative.

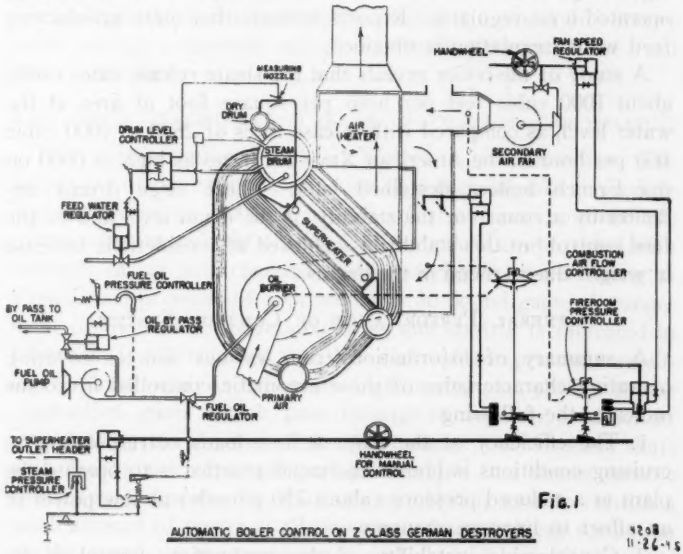


Fig. 1

AUTOMATIC BOILER CONTROL ON 2 CLASS GERMAN DESTROYERS

A separate adjustable cam on the cam shaft positions a spring to load a differential pressure type controller connected across the burner throat. This controller actuates the dampers in the windbox to regulate the flow of combustion air in accordance with the demand established by the cam and measured by the differential pressure connections. The cam shape is adjustable to provide the necessary relation between air and fuel at all loads.

A third adjustable cam is provided to load the spring of a fireroom pressure controller which actuates the steam valves of the secondary air fan turbines. This cam is likewise adjustable to provide the desired secondary air pressure at any load.

All regulators are provided with individual spring adjustments which permit alteration of standard conditions maintained.

Feed Water Control.

Regulation of the feed water is obtained by means of a diaphragm unit responsive to boiler drum level and to saturated steam flow as measured by a venturi in the steam line from the drum. This unit actuates a hydraulic pilot valve furnishing oil pressure to a double-acting piston which moves the feed water regulating valve. This is a conventional type two-element relay operated level regulator. Reports indicate that quite satisfactory feed water regulation is obtained.

A study of the boiler reveals that the steam release rate is only about 1000 cubic feet per hour per square foot of area at the water level, as compared with release rates of 2500 to 4000 cubic feet per hour in the American Navy and rates as high as 6000 on the French boilers described later. These large drums undoubtedly account for the stability of the drum level and of the feed control but this stability is achieved at considerable increase in weight due to metal in the drums.

General Performance of Control System.

A summary of information from various sources covering operating characteristics of these automatic controlled firerooms indicates the following:

1. The efficiency of the plant at low loads corresponding to cruising conditions is poor. The usual practice is to operate the plant at a reduced pressure (about 750 pounds) at low power in an effort to improve economy.

2. Considerable instability of the combustion control is reported at low rates of operation. This may be due to burners and furnace design or due to improper adjustment of the control.

3. Both destroyers are reported to be bad as far as smoking is concerned. This may be due somewhat to burner and furnace design and to the fact that satisfactory wide range burning equipment is not provided. However, reports indicate that the control of the secondary air fans and dampers is not well coordinated and that manual control of the fans is used at all times in an effort to keep smoking to a minimum. The arrangement of control which requires the fuel to be changed before a change in air is demanded leads to smoky combustion on increase in load.

4. The relation between fuel and air maintained by the control changes from time to time because of dirt in the fuel valve which alters its flow characteristic. Adjustments are provided for this condition but the fireman must be depended upon to establish proper combustion conditions.

5. Steam pressure is maintained with ± 20 to 30 pounds during normal maneuvering but manual control is occasionally resorted to for fast maneuvers.

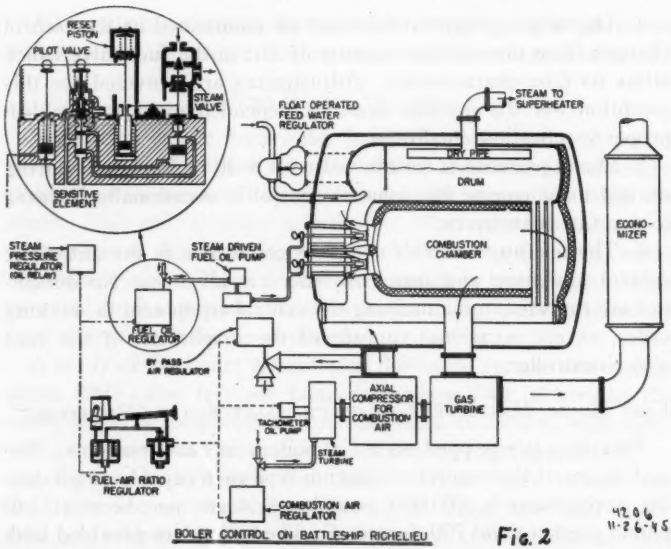
6. The operators exhibit complete confidence in the automatic control equipment and apparently use it at all times. No difficulties are reported in maintaining the control equipment in working order, except occasional rupture of the diaphragm of the feed water controller.

Fire Room Machinery—French Battleship "Richelieu".

This ship is equipped with six boilers in two firerooms. The boilers are of the natural circulation type each capable of producing approximately 300,000 pounds of steam per hour at 340 pound pressure and 700 degrees F. The boilers are provided with a double casing with combustion air entering the outer casing at a maximum pressure of approximately 30 pound gage. Arrangement of the boiler, boiler auxiliaries and control is indicated in Figure 2. It will be noted that the axial type compressor furnishing combustion air is driven by a gas turbine supplied with the combustion gases which pass through a separate economizer after leaving the exhaust of the gas turbine. On the same shaft with the axial compressor and gas turbine is a steam turbine which supplies power for starting and which is used to furnish a small amount of power at all times in order to achieve regulation of the speed of the compressor and thus the combustion air supply. Each boiler is equipped with four oil burners of the mechanical atomizing type taking fuel oil at a maximum pressure of approximately 430 psi. Each burner is capable of burning 5000 pounds of oil per hour.

Arrangement of Control.

The automatic control system consists of a steam pressure regulator which develops an oil pressure proportional to steam pressure. This oil loading pressure is connected to a reset or standardizing type regulator on the valve supplying steam to the steam driven fuel oil pump which supplies the burners. The same



BOILER CONTROL ON BATTLESHIP RICHELIEU Fig. 2

oil loading pressure is delivered to a fuel-oil ratio regulator to establish a proportional demand for combustion air. Flow of combustion air is indicated by a tachometer oil pump connected to the axial compressor shaft, the oil pressure developed by the tachometer being proportional to compressor speed. The tachometer oil pressure is connected to the fuel-air ratio regulator which permits adjustment of the delivered tachometer pressure in accordance with the load demand established by the steam pressure regulator and the number of burners in service and the excess air condition required.

The readjusted tachometer oil pressure is used to actuate the regulator on the steam valve to the auxiliary turbine on the axial compressor shaft to obtain a compressor speed and thus air flow in accordance with the demand. To obtain stability and suitable range of control the readjusted tachometer oil pressure is also connected to a regulator in a by-pass line from the discharge of the secondary air compressor so that the actual quantity of combustion air may be adjusted as well as the compressor speed.

Feed water regulation is obtained by means of a float-operated regulator with water-operated pilot for actuation of the feed water valve. This regulator is similar in design to the Weir regulator used extensively in the British Navy.

General Performance of Control.

From observation of the operation of the boilers and from reports received from different sources, the following conclusions are drawn:

1. Considerable difficulty was encountered with the feed water control on these boilers at high loads due to a bad surging condition in the boiler drum. As previously noted the release rate on these boilers is about 6000 cubic feet per hour per square foot of area at the water level. With the type of feed water regulator used, the regulation of feed water flow was quite unsteady due to this drum surge and considerable swinging of loads between boilers was encountered because of erratic water control. At low loads operation of the feed water control was satisfactory.

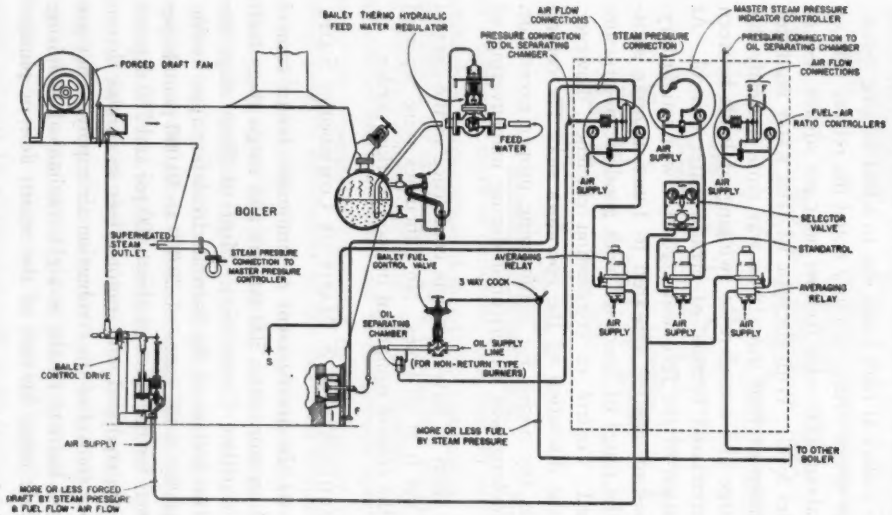
2. The combustion control appeared to function quite satisfactorily over the range of operation with good steam pressure being maintained. Control of combustion efficiency, however, was purely at the discretion of the operators since the control does not include any means for maintaining proper excess air. Considerable smoking was experienced during maneuvering of the plant.

3. Operators seem to have complete confidence in the control equipment and use it at all times. No difficulties were reported in maintaining the control equipment in good working order.

Fire Room Machinery—U. S. Maritime Commission Ships.

Figure 3 shows the arrangement of automatic boiler control equipment used on more than 500 tankers and cargo ships built by the U. S. Maritime Commission. Most of these ships are equipped with two boilers of the natural circulation type having a maximum capacity in the range of 20,000 to 50,000 pounds per hour and designed for operation at about 400 psi and 750 degrees F. The boilers are equipped with motor driven or turbine driven fans with damper control of the combustion air supply. They are provided with oil burners of the straight mechanical atomizing type or with wide range burners of the return flow or plunger types.

The combustion control system used is of the pneumatically operated type. The equipment consists of a steam pressure controller which develops an air pressure proportional to steam pressure. This air pressure is delivered directly to a diaphragm



DIAGRAMMATIC LAYOUT OF BAILEY COMBUSTION AND FEED WATER CONTROL SYSTEMS
FOR MARINE SERVICE

SUB No 555

BAILEY METER CO.
D504,884-A

Fig. 3

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motor valve in the fuel supply line to the burners and to a pneumatic piston operator for actuating the combustion air damper.

A metering type fuel-air ratio controller continuously determines the relation between fuel oil flow and combustion air flow and applies a readjusting air pressure to the averaging relay in the loading line to the secondary air damper control drive. In this system, therefore, the fuel and air supply to the furnace is changed simultaneously in accordance with steam pressure demands and the fuel-air relation is continuously readjusted by a metering type ratio controller.

Feed water flow to the boiler is regulated by a conventional self-acting thermo-hydraulic feed water regulator responsive to the level in the boiler drum.

General Performance of Control.

In view of the unprecedented shipbuilding program carried out in this country in the last few years, the application of automatic boiler control has come much faster in the Marine field than in the Stationary Power field. Shipbuilders, ship operators and manufacturers of control equipment were unfamiliar with this new device for improvement of Marine Power Plant operation. In spite of this fact, the equipment has served well and has gradually gained the complete confidence of the operators. Observations and reports from many sources lead to the following conclusions:

1. Automatic boiler control aids materially in the operation of a ship provided with inexperienced fire room personnel.

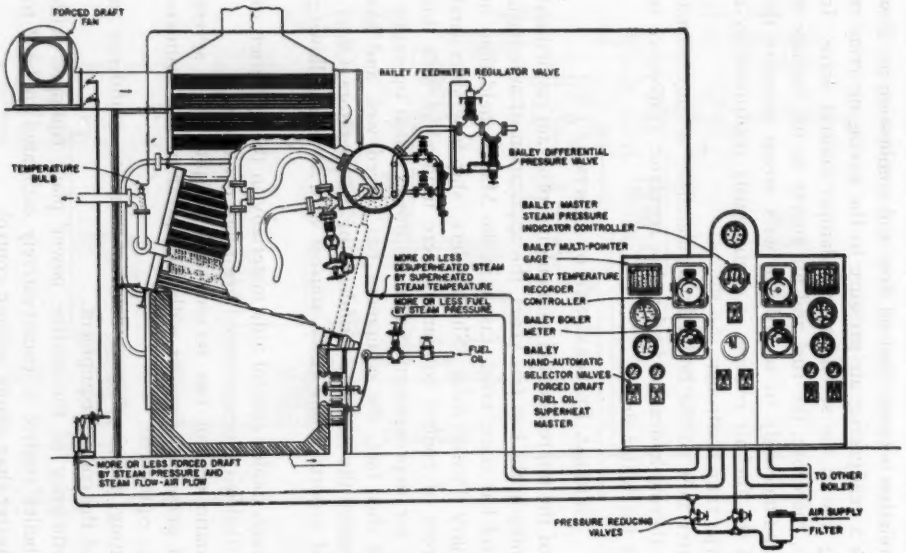
2. Automatic control has served well in spite of a severe shortage of personnel familiar with the operation and maintenance of this equipment.

3. Operating personnel soon develop complete confidence in operation of the control equipment.

4. The efficiency of the marine power plant equipped with automatic boiler control is conservatively estimated to be 2 to 4 per cent better than plants without control.

Fire Room Machinery—Atlantic Refining Tankers.

One of the early applications of automatic boiler control was to the turbo electric tankers built for Atlantic Refining Company prior to the war period. These ships are equipped with two water


FORCED DRAFT FAN
BAILEY FEEDWATER REGULATOR VALVE
BAILEY DIFFERENTIAL PRESSURE VALVE
TEMPERATURE BULB
BAILEY MASTER STEAM PRESSURE INDICATOR CONTROLLER
BAILEY MULTI-POINTER GAGE
BAILEY TEMPERATURE RECORDER CONTROLLER
BAILEY BOILER METER
BAILEY HAND-AUTOMATIC SELECTOR VALVES FORCED DRAFT FUEL OIL SUPERHEAT MASTER
MORE OR LESS DESUPERHEATED STEAM BY SUPERHEATED STEAM TEMPERATURE
MORE OR LESS FUEL BY STEAM PRESSURE
FUEL OIL
MORE OR LESS FORCED DRAFT BY STEAM PRESSURE AND STEAM FLOW-AIR FLOW
TO OTHER BOILER
PRESSURE REDUCING VALVES
AIR SUPPLY FILTER

DIAGRAMMATIC LAYOUT OF BAILEY METERING AND COMBUSTION CONTROL SYSTEM
FOR ATLANTIC REFINING COMPANY TANKERS
S.S. A.W. VAN DYKE
S.S. W.H. COLLEY

Fig. 4

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1-23-29

tube boilers each capable of producing approximately 25,000 pounds of steam per hour at 625 psi and 850 degrees F. steam temperature. Later ships were designed for 900 degrees F. steam temperature.

Arrangement of the boiler control system is indicated on Figure 4. The combustion controls and feed water regulators are similar to that described above with the following exceptions:

1. Recording type instruments are used throughout so that the engineers have a complete record of the performance of the power plant during the entire voyage.

2. The Steam Flow-Air Flow System is used for a combustion guide and the control of fuel air proportioning is readjusted from this Steam Flow-Air Flow relation.

3. In view of the high steam temperature, the boiler is arranged with primary and secondary superheaters with a steam attemperator located in the steam drum for desuperheating steam leaving the primary superheater and a by-pass valve automatically controlled for maintaining final steam temperature.

General Performance.

The following statement extracted from the paper by Mr. L. M. Goldsmith, Chief Engineer of Atlantic Refining Company, indicates the performance of automatic boiler control on these ships:

"It was possible to determine the output and efficiency of the boilers closely. The boilers were designed to give an efficiency of 85.5 per cent at the rate of load of 22,500 pounds per hour actual output . . . During the trial two fuel test runs were made, one of four hour and one of three hour duration. During the first of these runs, the boilers were operated at a pressure of 620 psi and a temperature of 855 degrees F. producing 21,800 pounds per hour each or equivalent evaporation of 25,100 pounds per hour . . . These results gave an efficiency of 86.6 per cent, stack loss of 11.3 per cent, and an unaccounted for loss of 2.1 per cent.

"The second run was at almost exactly the same rate and conditions showed an efficiency of 86.9 per cent with stack losses of 11.3 per cent and unaccounted for losses of 1.8 per cent.

"Since these tests were made, the results under normal operating conditions have shown that these efficiencies are easily maintainable. Data taken from the log of the homeward (loaded) leg of voyage No. 11 showed that an efficiency of 87.3 per cent was maintained for the four consecutive full days on the run.

We believe that such efficiencies can only be obtained and maintained by the almost perfect combustion conditions made possible by automatic combustion control".

Fire Room Machinery—U.S.S. "Dahlgren" DD187.

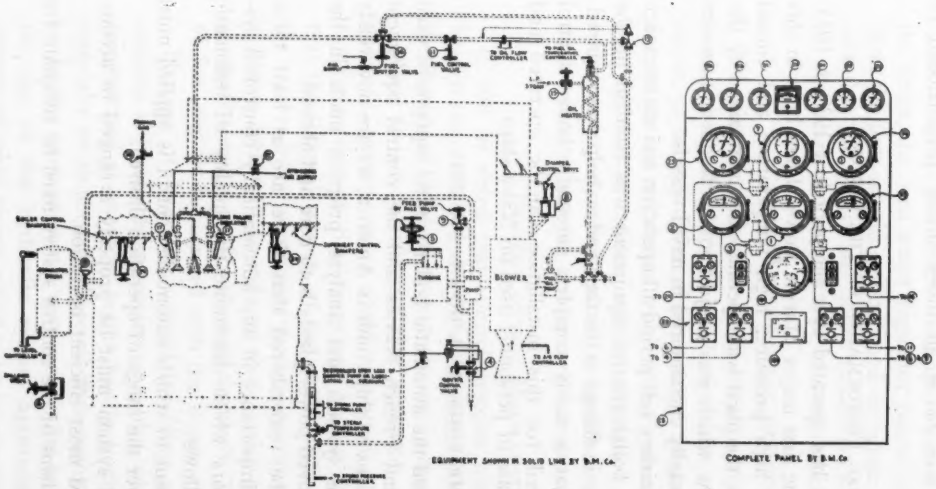
In 1937 the U. S. Navy selected one of the four stack destroyers of World War I vintage for test of a high pressure-high temperature power plant for propulsion purposes. One fire room of this destroyer was equipped with two forced circulation boilers of the Steamotive type suitable for generation of 120,000 pounds of steam per hour at 1200 psi and 900 degrees F.

The boilers were equipped with wide range steam atomizing oil burners and each boiler was provided with a combined auxiliary set consisting of the secondary air blower, the fuel oil pump, and a 10 cylinder reciprocating type feed pump all driven from a single speed variable steam turbine.

These boilers were provided with complete automatic boiler control as indicated on Figure No. 5. Arrangement of this automatic boiler control system and the necessary indicating and controlling instruments is shown on the diagram.

The speed of the combined boiler auxiliary set and thus the delivery of feed water from the reciprocating feed pump is dictated by the indicator (1) showing total steam output of the boiler. Readjustment of the set speed and thus water flow is obtained from the separator drum level indicator-controller (2). Fuel and air supply to the boiler changes simultaneously with feed water flow as established above. The combustion air supply is readjusted by a damper at the blower discharge actuated by the pneumatic power cylinder operated by the steam pressure indicator-controller (7). The proper relation between fuel and air is maintained by the ratio controller (10) which readjusts the fuel control valve (11) . The steam temperature is maintained by adjustment of the gas dividing dampers (24) from the steam temperature indicator-controller (23).

Flame failure indicators, automatic lighting torches and the automatic ignition control is at 17, 18 and 18-A on the diagram. Remote hand or automatic selector stations (20) are available at the control panel for manual control under unusual conditions. The annunciator (18-B) indicates to the operator any faulty operation of the units.



1. STEAM FLOW CONTROLLER
2. DRUM LEVEL CONTROLLER
3. DIFFERENTIAL RELAY
4. TURBINE GOVERNOR CONTROL VALVE
5. AUXILIARY TURBINE GOVERNOR
6. DRUM SPILL OVER VALVE
7. STEAM PRESSURE CONTROLLER
8. BLOWER DAMPER CONTROL DRIVE
9. FEED PUMP BY PASS VALVE
10. OIL FLOW-AIR FLOW CONTROLLER
11. FUEL OIL CONTROL VALVE
12. ATOMIZING AIR VALVE
13. EXCESS PRESSURE VALVE
14. FUEL OIL TEMPERATURE CONTROLLER
15. OIL HEATER STEAM VALVE
16. FUEL OIL SHUT OFF VALVE
17. FLAME FAILURE INDICATORS
18. AUTOMATIC BURNER LIGHTING EQUIPMENT
19. LOW WATER LEVEL THERMOSTAT SWITCHES
20. HAND-AUTOMATIC SELECTOR VALVES

21. PRESSURE GAGES
22. TEMPERATURE INDICATORS
23. STEAM TEMPERATURE CONTROLLER
24. SUPERHEAT DAMPER CONTROL DRIVES
25. INSTRUMENT PANEL

AUTOMATIC CONTROL
FOR
OIL FIRED STEAMOTIVE BOILER.

Fig. 5 2714 4/24/37

The control system is pneumatically operated and is of somewhat unusual design in view of the forced circulation boilers used. In the forced circulation boiler, it is, of course, essential that feed water flow be at all times closely proportioned to steam output.

General Performance of the Control.

This equipment has operated over the period of 1939 to 1945. While this was the first major Marine Power Plant built in this country utilizing high pressure, high temperature and forced circulation boilers, the plant was operated and maintained by the regular Navy crew which was subject to the frequent turnover of personnel ordinarily encountered in naval vessels.

After the preliminary test period all operation and maintenance of the automatic boiler control equipment was handled by the regular Navy crew in charge at the time and no help was required. This indicates that a fairly complex automatic boiler control system as required for this installation can be operated and serviced by the class of personnel used on Navy ships.

Comparison of Control Systems.

Having described the automatic boiler control equipment used in the German and French Navies and the control equipment used in various types of ships built in America, we are now ready to compare these systems and analyze progress made in the direction of improved power plants both here and abroad.

It must first be remembered that there are at least three factors of equal importance in any automatically controlled system. These factors which determine the quality of automatic control are as follows:

1. The equipment to which automatic control is applied must be controllable over the range of operation desired.
2. The control system must be properly arranged to accomplish the best and most efficient operation.
3. The mechanisms of the control system must be adequate for the job.

Controllable Equipment.

If the fuel burning equipment does not have a range of efficient performance suitable to the load requirements application of automatic control will not correct it. If combustion air

fans do not have a stable range of speed control sufficient for the load range, then supplementary control of air is required for efficient operation. If boiler circulation and steam release area is not adequate and severe surging of water level takes place in the drum, the feed water regulator cannot correct this. Therefore, the first step in getting good automatic control is to make sure that the boilers and boiler auxiliaries are of proper design for the range of load and type of load to be encountered.

Proper Arrangement of Control.

The control equipment must be properly arranged for the type of boiler and type of auxiliaries to be controlled. As noted above, an entirely different arrangement of the control is required for the forced circulation boiler with combined boiler auxiliaries than for the conventional natural circulation boiler system. The boiler control equipment must be flexible enough to adapt itself to the various general arrangements required.

Proper Control Mechanisms.

It will be noted that control systems used on German and French ships utilize hydraulic mechanisms whereas those used on the American ships use pneumatic equipment.

While it is acknowledged that hydraulic control, if properly designed, permits more rapid operation of the power units, experience in both Stationary and Marine Power Plants shows that the thermal storage of the steam boiler and the inertia of the various boiler auxiliaries permits operation of the control on a slower cycle (10 seconds or more) with greater stability and adequate speed to satisfy the load demands. Better proportioning of fuel and air is obtained with the slower cycle.

The use of hydraulic control involving oil as the actuating medium is justifiable in turbine governing where all oil piping is concentrated at one spot but is hardly justifiable in fire rooms especially on shipboard where oil lines running to various points in the fire room present a very serious fire hazard. Furthermore, the use of the hydraulic system approximately doubles the amount of small piping which must be run since drain lines must be provided to return the oil whereas with the pneumatic equipment these drains are unnecessary. These additional drain lines not only increase the cost and complication but still further increase the fire and operating hazard in battle.

Keeping the above bases of comparison in mind, we can now look at the various controls used.

German Destroyers.

The unstable combustion conditions at low rates is probably due to inferior design of the fuel burning equipment and application of control does not correct this condition. The basic arrangement of the control is not adequate since no means beyond the judgment of the operator is provided for insuring proper excess air control. Finally, while the hydraulic mechanisms are entirely adequate, there is no need for increasing the fire hazard when simpler and less hazardous type of mechanisms would perform equally as well. The basic arrangement of control which provides instantaneous changes in fuel without waiting for changes in blower speed causes smoking during rapid maneuvers.

French Battleship.

The general arrangement of boiler and auxiliaries on this installation appears to lend itself readily to automatic boiler control. It is doubtful that the oil burners give efficient operation over the entire range of load but reliable data in this respect is lacking. The arrangement of the automatic control system is not adequate since provision for fuel-air proportioning is even less effective than in the case of the German ships. The system depends entirely too much upon the judgment of the operators. As in the case of the German ships, the use of hydraulic control with the large number of oil pumps and oil piping required presents an entirely unnecessary fire hazard.

American Ships.

Considerable development is still required to achieve a completely satisfactory automatically controlled fire room. We are still lacking in means of burning fuel efficiently over a wide range. When wide range burners are available, automatic control of Marine Power Plants will be much improved.

The control systems used on American ships are flexible enough to adapt themselves to any type of boiler and boiler auxiliary arrangement. Likewise, the control equipment now in use is adequate. It is simple in design and free of fire hazards. Many improvements are continually being made in the control mechan-

isms and it is to be expected that this art will progress as fast as the design of burning equipment and of other boiler auxiliaries.

Electronic control systems have been used extensively in aircraft and fire control work and can be adapted to boiler control when they have reached a proper stage of development. At the moment, however, the pneumatic system is more adaptable to fire room conditions and is easier for the present class of firemen to understand, operate and maintain.

Developments in the U. S. Navy.

Many of the engineers in charge of design of naval craft are fully aware of the advantages of automatically controlled fire rooms and engine rooms on naval craft. However, they are faced with a problem peculiar to the naval power plant which must be overcome before automatic control is generally adopted.

Naval power plants must be designed for high capacities within small weight and space limits. They must be operated at a small fraction of their capacity during large proportion of their operating life. While the average mechant ship has a normal load range of only about 4 to 1 with a large proportion of the operating time at the high load, destroyer power plants must have a load range of almost 15 to 1 and must operate at about 10 per cent of maximum load for all but a small fraction of the operating time.

It is evident, therefore, that the naval power plant must be designed for efficient performance at low rates as well as high rates and must be capable of efficient maneuvering at any load and over any range of load. It is only natural, therefore, that naval engineers have delayed the adoption of the automatic controlled fire room until burning equipment and boiler auxiliaries were capable of control over the range required.

The Navy has been engaged in the development of wide range burners for a number of years and has made considerable progress. Figure 6 shows the performance of two burners now being tested. It is believed that each of these burners compares favorably as to range, excess air, and boiler efficiency with any fuel burners developed here or abroad.

It appears that either of these burners will permit efficient operation under complete automatic control over the load range required in destroyer service and will permit continuous operation at any load rate encountered with efficiencies equal to or better than that obtained with hand firing under ideal conditions.

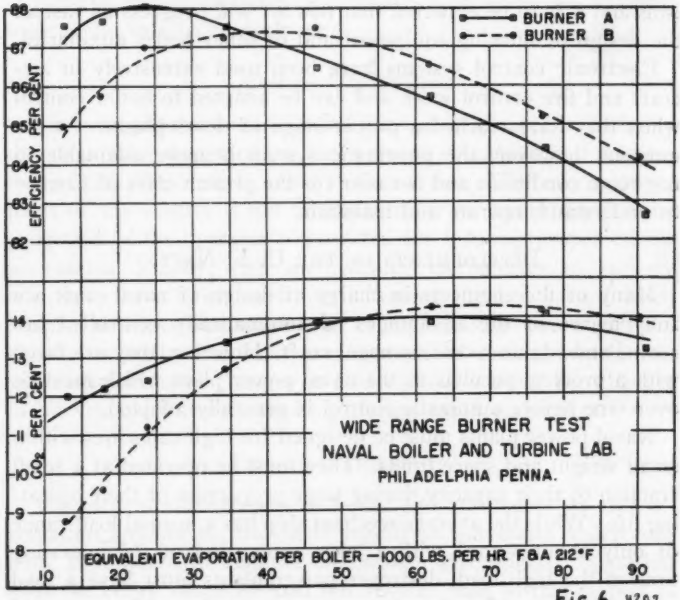


Fig. 6 4207 11-26-45

In view of the many other advantages available with automatic control and in view of the fact that the major problem of design is now being overcome, we believe that adoption of automatic boiler control will be seriously considered for future naval vessels.

CORDAGE, RESEARCH AND USE.

By G. I. Dewey* and W. P. Whitlock III†.

Cordage; which consists of rope and small stuff, has hundreds of diversified applications in the Navy, Figures 1 and 2. Its use is essential to the operation of the Fleet. In peacetime the supplies of fibers were plentiful and only the most suitable for the purpose were specified. Prior to the outbreak of World War II cordage for the Navy was manufactured of manila, flax, American hemp and cotton. When the Japanese struck at the Philippines in December 1941 they deprived the Allies of the source of the most desirable marine fiber, manila (abaca). Hitler had already cut off the principal sources of flax, which is grown in the European countries which were brought under Axis control.

Fortunately this situation insofar as manila is concerned had been foreseen and steps taken to alleviate it. In 1937 the United States Navy began stock piling manila fiber and many millions of pounds were in storage in warehouses in January 1942, at which time it was estimated supplies on hand were sufficient to carry through to the summer of 1943.

Several further steps were taken early in 1942 with a view to stretching the available supplies to the greatest extent possible. Among them was a program of conservation which included research and investigation under the Bureau of Ships of the numerous fibers which might have possibilities in manufacture. Also investigated were preservative treatments and changes in manufacturing processes the use of which might increase service life or produce greater quantities of cordage. This program was carried on principally at the Materials Laboratory and Ropewalk, Navy Yard, Boston, with cooperation of the Wm. F. Clapp Laboratories, Duxbury, Mass., and Industry.

In connection with enlarging supplies of cordage fibers, American hemp, jute, cabuya, istle, pita floja, caroa, ramie, Haitian

* Principal Engineer (Materials) Research and Standards Branch, Bureau of Ships, Navy Department, Washington, D. C.

† Lieutenant, U.S.N.R., formerly attached to the Bureau of Ships, Navy Department, Washington, D. C.

hemp, coir, henequen, Brazilian marinha yarns, and the blended sisal-hemp mixtures were evaluated for suitability as substitute fibers. The sisal-hemp mixture and jute were used to a large extent, mainly because they were available. Coir was used for fenders and cargo mats, istle as a sisal diluent, and henequen and jute in cargo nets. The other fibers were of little importance from the standpoint of supply and applicability.

The testing of synthetic fibers was also pursued, most work being done on nylon, which shows promise. It is strong, durable, and has a high elongation, and capacity for absorbing work, is resistant to fungi, but not as resistant to weathering.

In addition to the stock piling of manila the Navy had also given encouragement to a project for growing abaca in Central America. The plantings were started by the United Fruit Company, with plants brought from the Philippines by the Department of Agriculture. This project is now yielding substantial quantities of fiber, having been expanded with plantations in Panama, Guatamala, Honduras, and Costa Rica. Tests of Panama abaca at the Navy Yard, Boston in 1939 showed that cordage made therefrom was comparable in strength and durability to standard manila. The fiber however, was found to be somewhat "bold" or harsh. Production of cordage of fiber grown in Central America was started by the Industry in 1944. This production has proved somewhat weaker, breaking strengths being about 10 per cent less than in rope made from Philippine fiber. These differences are due mainly to differences in methods of producing fiber and are being investigated in current research.

Thus instead of one standard type of cordage, several types had to be used, and substitutions made both for general use and for many items of equipment. The many equipment specifications covering items such as cargo nets, grapnels, life floats, collision mats, and hammock fittings had for years specified manila. Accordingly, these items were changed to use substitute cordage in the types and sizes applicable to the use. In addition, the Navy Yard, Boston developed a new type of line term "composite" which consisted of manila yarns spun around yarns of substitute fiber, such as sisal, placed in the center of the strand, Figure 3. This development was applicable to the larger sizes used for hawsers. It saved large quantities of manila fiber and gave essentially the equivalent of manila in performance. This line

was supplied by Boston for Standard Stock and later on by Industry.

Not only "hard" fibers, such as manila and sisal, but also the "soft" fibers were critical in supply. Flax was used in the Naval service principally for signal halliards, shot lines, and fishing lines. When it became apparent that the war in Europe would shut off that source of supply and that the limited quantities grown in the United States, principally Oregon, would be inadequate an investigation of suitable alternates was authorized at Boston in 1940. Whereas several materials, including synthetic fibers and spun glass were evaluated, emphasis was placed on cotton as being readily available in large quantities. As a result smaller flax items were changed to cotton. Signal halliards, which constituted the largest use of flax, require resistance to hot stack gases, as well as weathering, flexing durability, and good breaking strength. A solution to the problem was found in the use of specially treated cotton halliard which has been in production since 1942.

In addition to the evaluation of various fibers indicated in the foregoing paragraphs work also was done on several lubricants, preservatives and constructions with a view to obtaining best use of all fibers. Early investigation showed the value of copper naphthenate as a preservative, except for its effect on flexing durability at the higher concentrations. Several other preservatives, and combinations thereof, have also been evaluated.

For untreated cordage, the type of fiber, size of line, and degree of exposure all affect the rate of loss of serviceability. Differences in biological deterioration have been noted in the various fibers, American hemp being the least resistant fiber. However, differences with size of line and conditions of exposure are far more important than differences in fiber. This was confirmed by direct observation at four advanced bases in the Pacific Ocean Area, Figures 4a-4b. When lines were properly stowed under cover no trouble occurred. Those coils which were found to be in a rotten condition had generally been subjected to severe exposure outdoors, and were frequently mud stained, Figures 5a-5b. Treated jute lines were always observed free of growth under similar conditions. The overall loss was not of a high magnitude, however, for two reasons: (1) Severe exposures last only for the initial stages of setting up an advanced base. (2) Line was too scarce for large quantities to accumulate in depots and decay.

For treated cordage, size and conditions of exposure are also major variables, the choice of treatment depending on the use of the line. While the danger of deterioration from mildew becomes less as the conditions of use improve and greater care can be given to handling and stowage of lines, nevertheless tests indicate that cordage life can be increased appreciably where weathering plays a major role. The question of treatment becomes an economic one, such as the balancing of ordinary weathering versus wearing and the added cost of the treatment.

With a view toward improved cordage performance, abrasion resistance tests have been conducted on machines of special design at the Boston Ropewalk, Figure 6. In these tests the sets of variables are (1) Revolutions per minute of the machine, specified as 77, 155, and 250 respectively, (2) Load applied in pounds, 7¾, 15-9/16 and 20¾ respectively, (3) Length of arc (Inches) in which abrasion occurs, 5⅜, 10⅞ and 14-1/16 respectively, (4) Size of yarns designated as A, B, C, and D (of decreasing size) and (5), Position in the machine, designated as 1, 2, 3, 4, 5.

Individual readings are quite variable, averaging in range ± 50 per cent for each 10 readings for a given condition (108 conditions). The effect of position in the machine is large, position #1 being 30-40 per cent high, and position #5 being up to 50 per cent below the average. The effect became less as the investigation progressed, being roughly half of that at the start. In all probability this is indicative of the wearing of the steel abrading edge.

The load and length abraded have but minor effects on the position factor and variation. Yarn A (the largest) was least variable. Yarn D (the smallest and softest yarn) was most variable.

At 77 Rpm. the data can be approximated by the relationship.

$$\text{Revs. to failure} = \frac{280 \times 10^6}{(\text{load})^{1.7}\,(\text{ft./lb.})^{1.10}\,(\text{length abraded})}$$

Overall the equation is of similar form, with varying constants for each speed, and approximates

$$\text{Revs. to failure} = \frac{\text{a constant}}{(\text{load})^{1.5}\,(\text{ft./lb.})^{1.25}\,(\text{length abraded})}$$

In general then, the abrasion resistance is most severely affected by the load (or pressure) while lighter, softer yarns are more quickly broken.

Reports of nearly 3000 inspection tests have been put on IBM cards, and averaged. Sisal, manila, jute, and henequen ropes are summarized below for several manufacturers, and periods, in view of the interest in actual breaking strengths and comparisons represented in the data.

The comparison of strengths, uncorrected for slight variations in weight per foot and turn is as follows:

	Manila	*Sisal*	*Jute*	*Henequen*
No. Samples.....	411	1898	558	20
Ave. Strength/Manila Spec.	1.128	.942	.799	.649
Ave. Strength/Manila Spec.	1.00	.83	.71	.58
Specified Strength ratio....	1.00	.70–.80	.60	.60

The effect of time on sisal strengths shows a slight decline due to use of diluents such as istle, low grade sisal, sisal tow, and American hemp.

Period	*Dates*	*Sisal Strength Per cent*
1	Jul.-Dec. 1943	100
2	Jan.-Jun. 1944	97
3	Jul.-Dec. 1944	97

The variation of strength between various manufacturers averages ± 8 per cent. This value does not account for weight and turn differences which may alter the conclusion.

On preliminary examination, the effect of turn or degree of twist is not clearly defined in the data reviewed.

MANILA—AVERAGE (ALL FIRMS AND PERIODS).

Size Ins.	*Circ.*	*Turn*	*Lbs./ft.*	*Strength*	*Strength Manila Spec.*	*No. Samples*
5/8	.634	.70	.014	547	1.218	5
3/4	.770	.....	.018	703	1.172	3
1	.989	.795	.027	997	.997	10
1 1/8	1.16	.....	.038	1540	1.142	5
1 1/4	1.28	.....	.052	2000	1.143	2
1 1/2	1.62	.....	.083	3500	1.320	1
1 3/4	1.78	.....	.104	3700	1.072	2
2 1/4	2.28	2.11	.163	6330	1.173	13
2 1/2	2.50	2.32	.183	7250	1.116	87
2 3/4	2.69	2.51	.212	8187	1.063	10
3	2.89	2.78	.253	9780	1.089	50
3 1/4	3.16	2.95	.289	11400	1.086	12
3 1/2	3.43	3.06	.361	14710	1.227	4
3 3/4	3.70	3.24	.384	14310	1.060	10
4	3.93	3.55	.453	16150	1.077	48
4 1/2	4.36	3.74	.564	22900	1.238	11
5	4.98	4.31	.718	25300	1.123	92
6	5.81	5.08	1.060	36300	1.171	6
7	7.07	5.72	1.360	43470	1.060	3
8	8.03	6.78	1.860	57900	1.113	36
9	8.64	7.10	2.180	64000	1.000	1
					Average 1.128	Total 1411

SISAL—AVERAGE (ALL FIRMS AND PERIODS).

Size Ins.	*Circ.*	*Turn*	*Lbs./ft.*	*Strength*	*Strength Manila Spec.*	*No. Samples*
¾	.793	.815	.0185	648	1.080	90
1	.994	1.050	.0282	1036	1.036	128
1⅛	1.138	1.205	.0380	1403	1.040	80
1¼	1.273	1.282	.0481	1713	.978	184
1½	1.545	1.670	.0707	2600	.981	447
1¾	1.747	1.707	.0928	3070	.890	53
2	2.03	1.982	.1253	4025	.915	194
2¼	2.29	2.335	.1510	5220	.968	94
2½	2.44	2.36	.1866	5760	.887	91
2¾	2.69	2.58	.213	6950	.903	13
3	2.94	2.76	.259	8200	.912	30
3¼	3.21	2.90	.313	9620	.917	21
3½	3.46	3.19	.346	12020	1.001	307
3¾	3.68	3.38	.411	12200	.904	22
4	3.91	3.58	.452	13370	.891	50
4½	4.50	4.02	.568	15920	.860	8
5	4.81	4.27	.719	21200	.941	41
6	5.93	5.00	1.020	28800	.928	13
8	7.89	6.35	1.875	45300	.870	32
				Average	.942	Total 1898

JUTE—AVERAGE (ALL FIRMS AND PERIODS).

Size Ins.	*Circ.*	*Turn*	*Lbs./ft.*	*Strength*	*Strength Manila Spec.*	*No. Samples*
¾	.775	.765	.0187	520	.867	31
1	.995	.983	.0272	727	.727	21
1½	1.55	1.57	.072	2095	.790	10
3	2.95	3.08	.253	7280	.810	496
				Average	.799	Total 558

CUBAN HENEQUEN—AVERAGE.

Size Ins.	*Circ.*	*Turn*	*Lbs./ft.*	*Strength*	*Strength Manila Spec.*	*No. Samples*
2	1.83	2.06	.116	3100	.705	5
2¼	2.10	2.23	.132	3590	.665	5
3	2.86	2.87	.242	5445	.605	5
3½	3.46	3.16	.356	7420	.619	5
				Average	.649	Total 20

An extensive exposure and preservative investigation has been underway to determine the rate of deterioration of lines, exemplified by Figures 7 and 8, and effectiveness of known preservation methods. This project has several parts:

(A) Large scale exposures designed to show the general deterioration of cordage under three conditions, in two latitudes, as measured in two ways. At the same time the effect of fiber treatment and size of line is being determined. The project continues in cooperation with the Wm. F. Clapp Laboratories and Navy Yard, Boston.

The variables are:

Exposure conditions: Soil Burial, Humid Stowage (under Tarpaulin) Sea Weather (Sun at Low Tide, Sea at High Tide).

Latitude: Duxbury, Mass. and Miami, Florida

Fiber: Manila, sisal, sisal-hemp, jute, American hemp

Size, (Circ): ½ inch, 1½ inch, 6 inch.

Treatment: None, Ammoniacal Copper, Copper Naphthenate, Chlorinated Phenol types.

Evaluation: Strength, Flexing Durability.

(B) Screening Tests of Preservatives at Boston.

The method uses small size three ply sisal twine, tested for resistance to soil burial, heat aging, chaetomium globosum inoculation.

(C) Evaluation of Manufacturers suggestions—Soil Burial.

(D) Weathering exposures of Twines at New Orleans, and Panama.

The results of this investigation are summarized below in terms of the number of days to reach 75 per cent of original strength. The half strength periods are available for only part of the investigation, and are not included as exposures are still underway.

¾ Strength Period—Days

Effect of Size:		½"	1½"	6"
Duxbury	Soil Burial....	3	9	10
	Sea-Weather..	190	165	..
	Humidity.....	270	270	..
Miami	Sea-Weather..	18	60	..
	Humidity.....	...	130	..

FIGURE 1.—AN "AKN" DRY DOCKING IN THE SOUTH PACIFIC. LARGE LINES ARE REQUIRED FOR SUCH OPERATIONS.

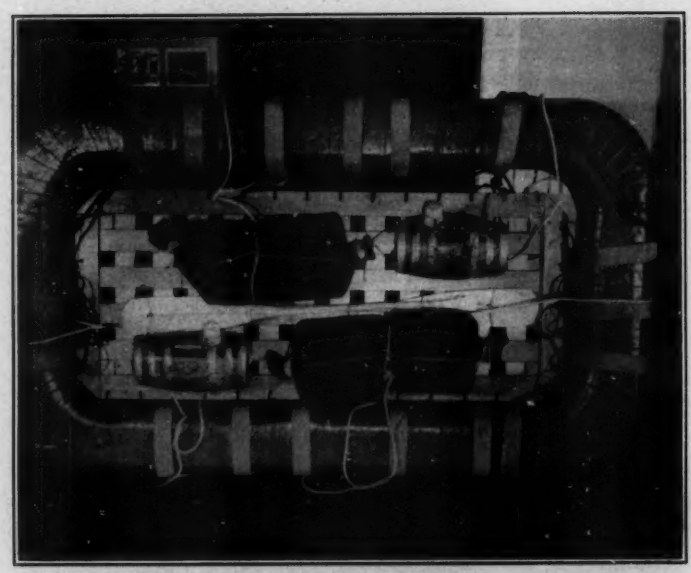


FIGURE 2.—LIFE FLOAT—THE USE OF SMALL LINES IS ESSENTIAL ABOARD SHIP.

FIGURE 3.—COMPOSITE ROPE.

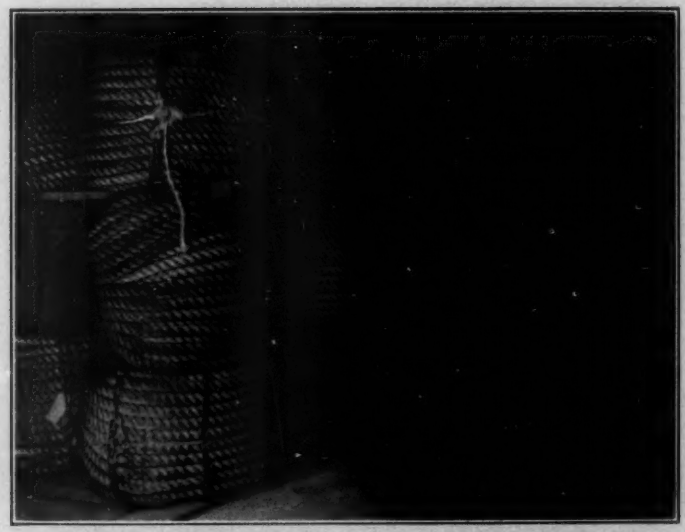


FIGURE 4A.—SOUTH PACIFIC BASE. GOOD STORAGE CONDITIONS.

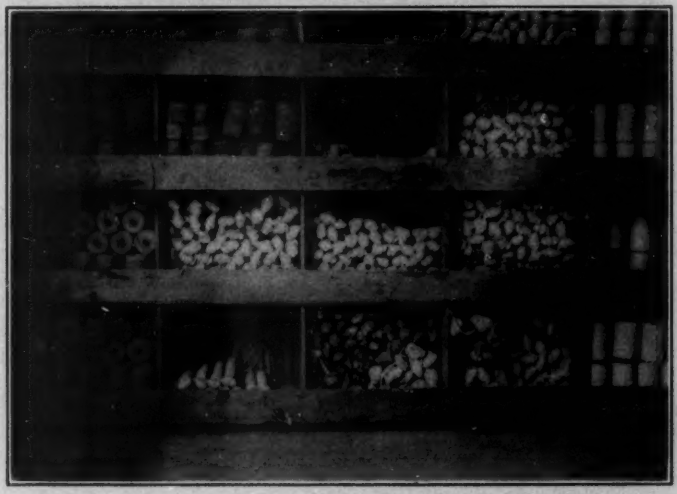


FIGURE 4B.—SOUTH PACIFIC BASE. GOOD STORAGE CONDITIONS.

FIGURE 5A.—SOUTH PACIFIC BASE. TEMPORARY STORAGE.

FIGURE 5B.—SOUTH PACIFIC BASE. POOR STORAGE.

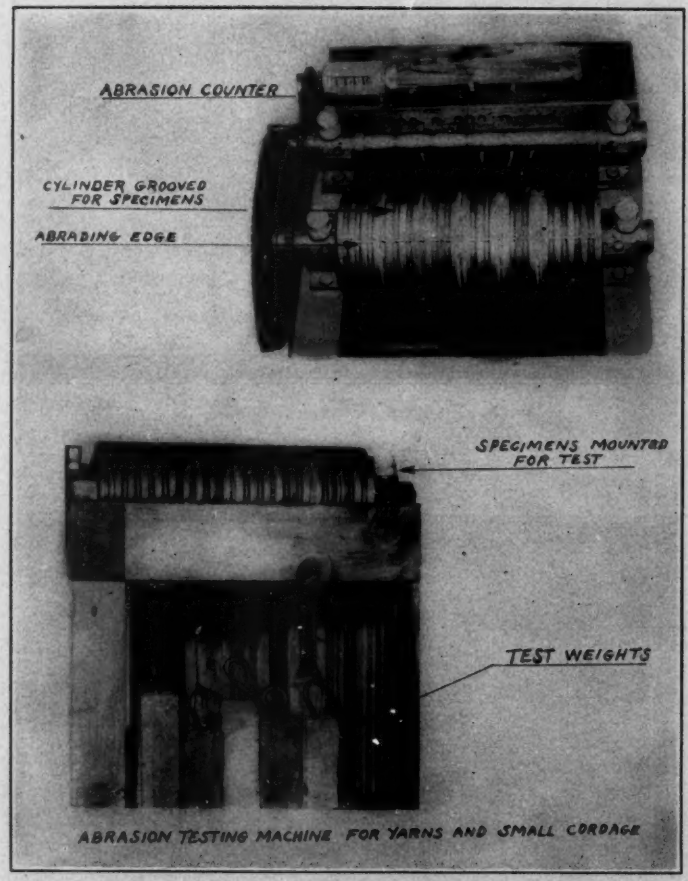


FIGURE 6.—ABRASION TESTING MACHINE.

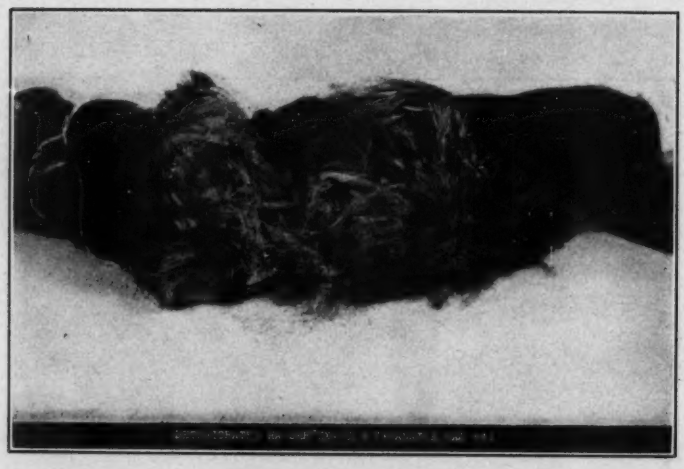


FIGURE 7.—DETERIORATION.

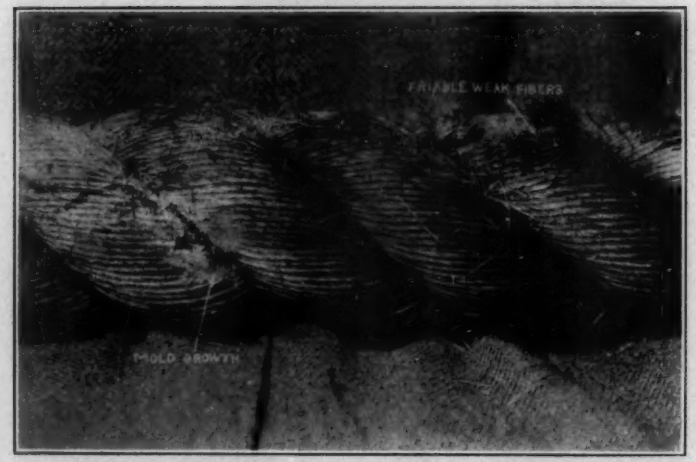


FIGURE 8.—DETERIORATION OF CORDAGE FIBERS.

Effect of Fiber:		*Manila*	*Sisal*	*Sisal-Hemp*	*Jute*	*Hemp*
Duxbury	Soil Burial....	13	9	5	11	3
	Sea-Weather..	200	165	125	200	140
	Humidity.....	190	270	175	300	150
Miami	Sea-Weather..	60	60	25	60	15
	Humidity.....	170	130	70	140	15

Effect of Treatment:		*None*	*Ammoniacal Coppers*	*Copper Naphthenates*	*Chlorinated Phenols*
Duxbury	Soil Burial....	9	70	160	80
	Sea-Weather..	165	280	240	160
	Humidity.....	270	280	160	300
Miami	Sea-Weather..	60	60	160	90
	Humidity.....	130	170	...	300

Comparison of Variables:

The average resistance of the various fibers under all conditions is in the following descending order of merit:

Manila
Jute
Sisal
Sisal-hemp
American hemp

The average performance of preservatives is in the following descending order of merit:

Copper naphtenate types
Chlorinated phenols
Ammoniacal copper types

The average order of severity of test is:

¾ STRENGTH

Condition:		*Untreated Days*	*Treated Days*	*Effectiveness of Treatment*
Duxbury	Soil Burial....	10	100	10x
Miami	Sea-Weather..	45	100	2x
Miami	Humidity.....	105	220	2x
Duxbury	Sea-Weather..	165	230	1½x
Duxbury	Humidity.....	240	250	1x

The average effect of test measurement is:

¾ PERIOD—DAYS

Soil Burial:	*Untreated*	*Treated*	*Effectiveness of Treatment*
Strength.....	10	100	10x
Flexing Durability.....	5	35	7x

Conclusions.

In the past, manila was the only fiber used for general shipboard cordage. Small amounts of soft fiber goods—marline, ratline, cotton lines, flax halliards, etc. were also used. Manila is the strongest and most durable cordage fiber. Yet, during the War, ships were successfully equipped with lines with the use of only a small proportion of manila. Therefore, it is considered that the most important conclusion of all the experiences, is that while manila is the best fiber for marine rope the less desirable and less durable fibers, such as sisal properly prepared and treated, can also do most of the tasks for which manila is preferred. In general when such substitution is necessary, sisal lasts about 50 to 70 per cent of the manila service life, depending on use, in contrast to the former belief that *only* manila could do the job. Of greater importance than fiber type is the conditions of use, methods of use, and choice of the size of line.

Similarly, cotton satisfactorily replaced flax in signal halliards and small stuff, even though weaker. In the signal halliards, the cotton was treated with a high percentage (25 per cent) of preservative lubricant, resulting in a cheaper but entirely satisfactory halliard. Jute was used in cargo nets, with a large saving of hard fiber. Specifications for the jute rope also required incorporation of mildew preservative compound to insure longer service life.

The question of preservative treatment of cordage obviously becomes less important in peace time when cordage is available in greater quantities and is handled with more care. However, because of the likelihood that Navy cordage will in the future be stocked and used in more areas where mildew deterioration would be encountered than heretofore, it would appear that mildew proofing and preservative treatment will be useful as a precautionary protective measure, for cordage be it manila or sisal. Such treatment, even if applied in dilute concentration with cost not to exceed 5 per cent of the cordage cost may, from available information, extend the life of cordage by about 25 per cent.

It will be of interest to note in conclusion that whereas it was estimated early in the war that the supplies of manila fiber on hand in the stock pile would be exhausted by the summer of 1943, that as a result of the diligent efforts of all hands concerned to effect conservation, by substitution, research, and development there was still a small amount of manila fiber available for manufacture of rope when the war ended two years later.

AIR LEAKAGE TESTS ON AIR-ENCASED BOILERS.

By Fred J. Wiegand* and James W. Murdock†.

Introduction.

Air leakage of boiler casing panel joints, never a negligible consideration in the design and operation of Naval Express boilers, became increasingly important with the advent of air-encased steam generators. Units of this type were first employed in the *Sampson* (DD394) Class destroyers. From that time until the present, Naval boiler air casings have been constructed of corrosion-resistant steel plates ranging from about 0.015 inch to 0.031 inch in thickness. Use of these thin plates made true tightness more difficult to obtain than would have been the case with thicker cross sections and in some instances it proved necessary to resort to the use of plastic casing sealing compounds used in conjunction with the asbestos rope packing normally employed. While absolute tightness is never obtained on boiler casings of this type, especially after long continued operation, the leakage in new boilers should not exceed 3 per cent of the quantity of air required for boiler operation. The necessity of minimizing air leakage and of determining its quantity in test boilers rendered it desirable to establish a list of certain features beneficial in securing relatively tight casings and to develop a method for test purposes whereby casing leakage could be ascertained with reasonable accuracy. Both the features considered conducive to casing tightness and the method used by the Laboratory in the determination of air leakage are set forth below:

(a) Panels should be as large as possible without preventing their removal through hatches, watertight doors, and similar shipboard openings. The number of panel joints should be kept to a minimum.

(b) Panels should be adequately reinforced to insure rigidity, thus reducing strain at the joints due to buckling or expansion.

(c) Clamps for removable casing doors should be strong

* Marine Engineer, Naval Boiler and Turbine Laboratory.
† Associate Mechanical Engineer, Naval Boiler and Turbine Laboratory.

enough to allow these doors to be drawn up sufficiently so that reasonable airtightness is obtained.

(d) Asbestos gaskets should be employed at all casing joints.

(e) Casing panels should be welded together at the joints wherever practicable to do so.

(f) Expansion joints should be provided where the casing adjoins steam drums, water drums, and similar parts of the boiler structure.

(g) Suitable air seals should be used where openings in the casing are necessary for piping and soot blowers.

Air Leakage of Boiler Casings.

The method of testing boiler casings for air leakage as used by this Laboratory is as follows: Air is supplied to the boiler from the Navy Yard compressed air system through a 5 inch i. d. pipe arranged as shown in Figure 1. An impact-flow nozzle is used as the air measuring device; it is either 2 inches or 3 inches in diameter depending upon the maximum flow required. Large Naval boilers require the use of the 3 inch nozzle while air leakage of small boiler casings can be measured with the 2 inch size. This type of nozzle combines the best features of the ordinary flow nozzle and the pitot tube method of measuring flow. The air is forced into the cylindrical throat section and the coefficient of contraction is thus unity for all practical purposes. The impact tube is at the exact center of flow and, since the velocity across the throat section is nearly uniform, the usual pitot tube correction becomes almost unity. By taking the static pressure at the throat section, the necessity for making a correction for the adiabatic expansion of air is also eliminated. The tests are conducted at ordinary temperatures so that the air temperature in the throat is therefore very close to that measured before the nozzle. The result of these construction features of the nozzle is to give a quick and reasonably accurate flow calculation involving only two pressures and one temperature measurement. The ratio of the nozzle diameter to the pipe diameter can be neglected since the total or impact pressure is used, thus eliminating the usual velocity of approach factor. Because the flow is independent of the ratio of the nozzle size to the pipe size, a shorter length of straight pipe before and after the nozzle can be used than is usual in flow measurement. This is particularly advantageous on shipboard tests where space is limited. The accuracy of these

SET-UP FOR INNER CASING LEAKAGE

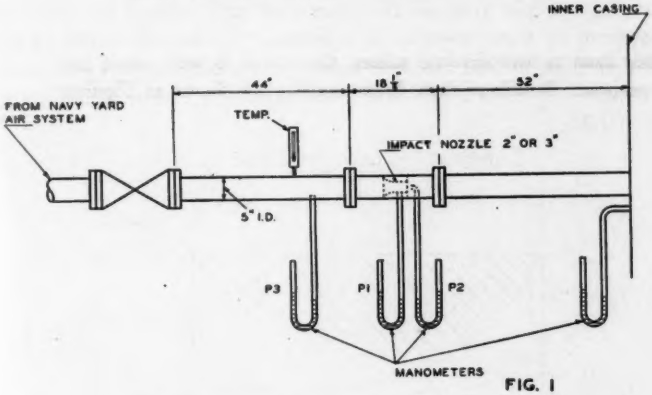


FIG. 1

3" IMPACT-FLOW NOZZLE

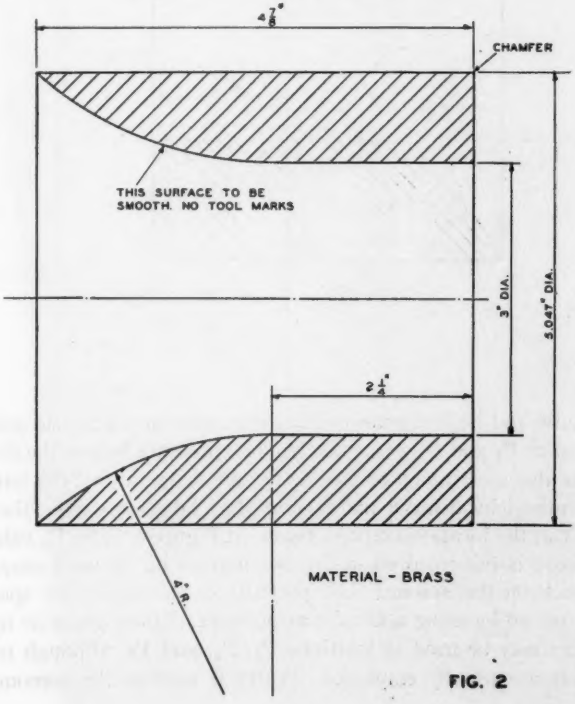


FIG. 2

simplified calculations has been found to be such that the error is within one per cent of the true flow rate. Since the primary purpose of these tests is to determine the fitness of the casing rather than to find absolute values, the nozzle is well suited for its purpose. Details of these brass nozzles are shown in Figures 2 and 3.

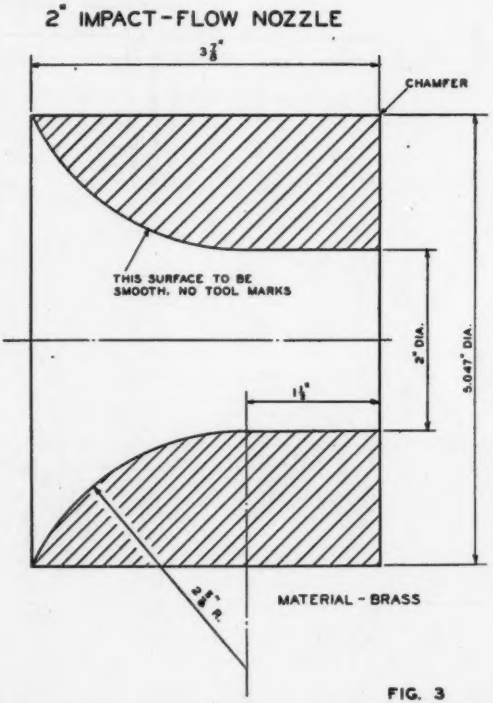


FIG. 3

Static and impact pressures are measured in the nozzle and are shown as P_1 and P_2, respectively. The pressure before the nozzle, P_3, is also ascertained as well as the air temperature; the latter is determined by using a mercury-in-glass laboratory type thermometer at the location marked Temp. in Figure 1. The P_3 value of pressure is not required in the calculations but is used simply as a check on the system. The pressure in the casing air space is determined by using a U-tube manometer. Either gages or manometers may be used at locations P_1, P_2, and P_3, although manometers are usually employed. Water is used in the manometers

for pressures up to about forty inches of water, The range of any of these manometers can be increased by replacing the water with carbon tetrachloride. For air pressures above sixty inches of water, mercury is used as the manometer liquid. The quantity of air passing through the test inlet pipe necessary to maintain a steady pressure in the boiler casing is equal to the amount leaking from the casing. It is determined by calculations as shown below.

The following tests are generally conducted:

(a) Leakage of inner casing after necessary caulking with a suitable boiler casing compound, but before bricking the furnace. This test is made before the outer casing is installed. The air inlet pipe is inserted into one of the burner opening closure plates. The remaining burner openings are tightly blanked and the uptake is also closed. A ½ inch thick asbestos gasket is used under the uptake closure plate. In addition, all openings in the casing for peep-holes, brick anchor bolts, etc., are plugged. The test is conducted at low pressures (not above ten inches of water) to prevent damage to the casing.

(b) Leakage of the outer casing only. This test and the following are conducted after the boiler brickwork is in place. The leakage of the outer casing is determined by blanking the uptake and burner openings as in the preceding test. In addition, the combustion air inlets must be blanked as shown in Figure 4. Also, soot blower air cooling lines and thermocouple holes are plugged and valves on the boiler air pressure test lines and the bottom casing drain must be closed. This arrangement permits leakage to occur only through the outer casing. The leakage through the

SET-UP FOR OUTER CASING LEAKAGE

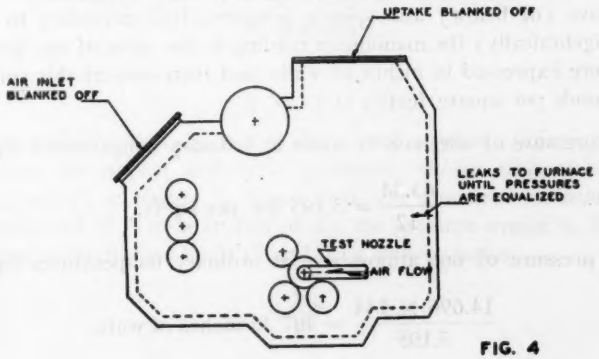


FIG. 4

inner casing will build up a pressure in the furnace until such pressure equals that between the inner and outer casings.

(c) Leakage of the inner and outer casings. This is determined by using the same set-up shown in Figure 4 except that the uptake is left open. This permits leakage of both casings. The leakage of the inner casing only, therefore, is obtained by subtracting the leakage of the outer casing from that of both casings.

The formula used by the Naval Boiler and Turbine Laboratory for calculating the air leakage is developed as follows:

The general equation for a perfect gas is

$$PV = WRT,$$

where

P = Absolute pressure of gas—lbs. per sq. ft.
V = Volume of gas—cu. ft.
W = Weight of gas—lbs.
R = Gas constant — 53.3 ft.—lbs. per lb. °T for air at ordinary temperatures
T = 459.7 + t
t = Temperature of gas—°F.

Transposing and substituting for R and T, the density of air is given by either side of the equation

$$\frac{W}{V} = \frac{P}{53.3\ (459.7 + t)} \qquad (1)$$

Since the subject air pressures are measured in inches of water above (or below) atmospheric pressure, it is necessary to add (algebraically) the manometer reading to the value of one atmosphere expressed in inches of water and then convert this sum to pounds per square feet.

A pressure of one inch of water at ordinary temperatures equals

$$\frac{62.34}{12} = 5.195 \text{ lbs. per sq. ft.}$$

A pressure of one atmosphere at ordinary temperatures equals

$$\frac{14.696 \times 144}{5.195} = 407.36 \text{ inches of water}$$

Therefore,

$$P = 5.2\ (407.36 + P_1),$$

where

P_1 = Static air pressure—inches of water

Substituting in the right-hand member of equation (1),

$$\text{Density of air, } D_a = \frac{5.2\ (407.36 + P_1)}{53.3\ (459.7 + t)} \text{ lb. per cu. ft.} \qquad (2)$$

The basic equation for flow of air through the convergent nozzle employed is

$$Q = 3600\ VAD_a,$$

where

Q = Mass flow—lbs. per hr.
V = Velocity of air—ft. per second
A = Area of exit section—sq. ft.
D_a = Density of air—lbs. per cu. ft.

The coefficient of discharge is assumed to be unity. The velocity, V, is determined from the equation

$$H = \frac{V^2}{2g},$$

where

H = Velocity head—ft. of air
g = Acceleration of gravity—32.2 ft. per sec. per sec.

Transposing,

$$V = \sqrt{64.4H},$$

or

$$V = 8.025\sqrt{H} \qquad (3)$$

The velocity head, H, is that due to the algebraic difference between the impact and static pressures, shown respectively as P_2 and P_1 in Figure 1. Since these are measured in inches of water and H is to be in feet of air, the pressure available, P_2-P_1, must be multiplied by 1/12 and by a density factor,

$$\frac{D_w}{D_a},$$

where

D_w = Density of water—lbs. per cu. ft.
D_a = Density of air—lbs. per cu. ft.

At the air temperatures prevailing during the subject casing leakage tests, D_w is considered to be 62.34 and D_a is found by equation (2) above. Therefore,

$$H = \frac{(P_2 - P_1)\, D_w}{12 D_a},$$

or

$$H = \frac{5.2\ (P_2 - P_1)}{D_a} \text{ feet}$$

Substituting in equation (3),

$$V = 8.025 \sqrt{\frac{5.2\ (P_2 - P_1)}{D_a}},$$

or

$$V = 18.3 \sqrt{\frac{P_2 - P_1}{D_a}} \text{ feet per second}$$

The exit area of the nozzle, A, is calculated by

$$A = \frac{\pi d^2}{4 \times 144},$$

where

d = Diameter—inches

Simplifying,

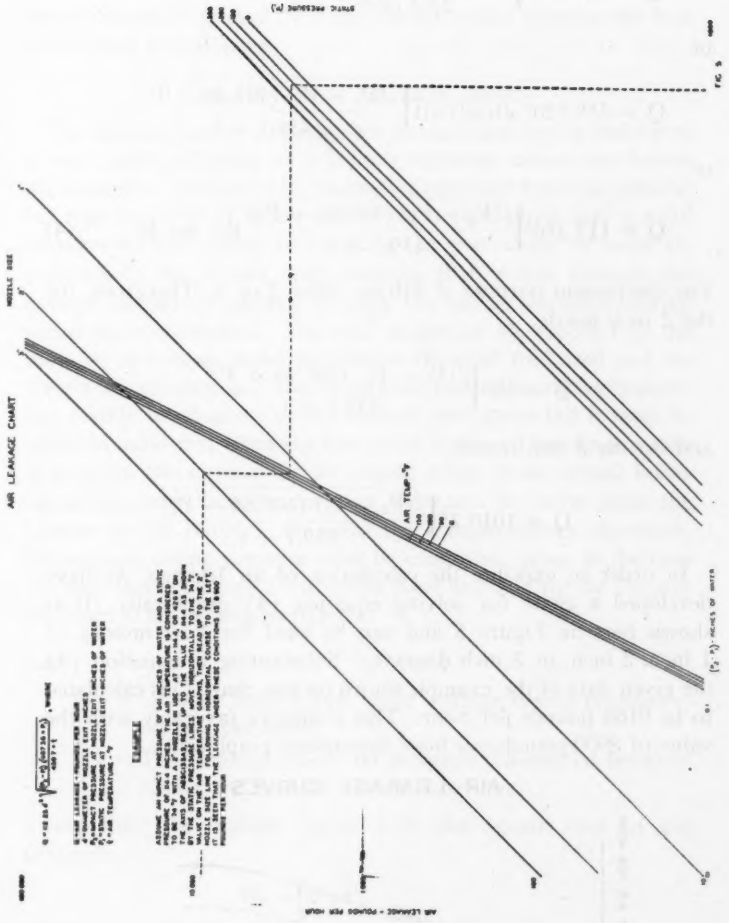
$$A = 0.00545417 d^2 \text{ sq. ft.}$$

Substituting in the basic flow equation,

$$Q = 3600 \times 18.3 \sqrt{\frac{P_2 - P_1}{D_a}} \times 0.00545417 d^2 D_a,$$

or

$$Q = 359.32 d^2 \sqrt{(P_2 - P_1)\, D_a},$$


AIR LEAKAGE CHART
AIR TEMP. °F
AIR LEAKAGE - POUNDS PER HOUR
EXAMPLE

or

$$Q = 359.32d^2\sqrt{\frac{(P_2 - P_1)\ 5.2\ (407.36 + P_1)}{53.3\ (459.7 + t)}},$$

or

$$Q = 359.32d^2\sqrt{0.097561}\sqrt{\frac{(P_2 - P_1)\ (407.36 + P_1)}{459.7 + t}},$$

or

$$Q = 112.25d^2\sqrt{\frac{(P_2 - P_1)\ (407.36 + P_1)}{459.7 + t}}\ \text{lbs. per hr.} \qquad (4)$$

For the present purpose, d will be either 2 or 3. Therefore, for the 2 inch nozzle,

$$Q = 449\sqrt{\frac{(P_2 - P_1)\ (407.36 + P_1)}{459.7 + t}}$$

and for the 3 inch nozzle,

$$Q = 1010.25\sqrt{\frac{(P_2 - P_1)\ (407.36 + P_1)}{459.7 + t}}$$

In order to expedite the calculation of air leakage, we have developed a chart for solving equation (4) graphically. It is shown here as Figure 5 and can be used for flow nozzles of 1 inch, 2 inch, or 3 inch diameter. Substituting in equation (4) the given data of the example shown on that chart, Q is calculated to be 9168 pounds per hour. This compares favorably with the value of 8900 pounds per hour determined graphically.

AIR LEAKAGE CURVES

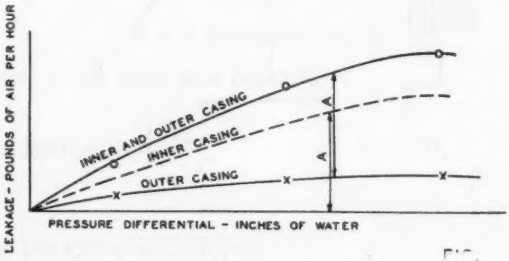


FIG.

Since the inner casing leakage after bricking the boiler is not measured directly, it must be determined by subtraction as mentioned above. This is done graphically as shown in Figure 6. The data from the leakage tests are plotted and a third curve, shown dotted, is plotted by using the difference between the first two curves as ordinates.

Boiler Casing Leakage Ratio.

The actual quantity of casing air leakage does not in itself give a very good indication of a boiler's tightness unless one knows the amount of leakage to be reasonably expected from the particular type and size of boiler tested. It is considered that a valid measure of the casing tightness is the percentage of total air supplied by the forced draft blowers that is lost through the casings instead of passing through the burner registers during actual boiler operation. The total weight of air supplied by the blowers, of course, is the product of the total fuel fired and the weight of air used per unit weight of fuel. The casing leakage can readily be obtained by the method here given but it must be borne in mind that when the leakage is measured the air pressure is uniform throughout the air casing while under actual boiler operating conditions the pressure decreases in value from the blowers to the burners. In using the leakage curves, therefore, the average casing pressure must be employed. Also, in the case of inner casing leakage a correction must be made for the furnace pressure. This is done as follows:

Let

W_1 = Leakage from curve, using average casing pressure—pounds per hour.

W_2 = True leakage based on pressure differential between casing and furnace—pounds per hour.

Then, since the leakage varies with the square root of the pressure,

$$W_1 = k\sqrt{P_c},$$

where

P_c = Average casing pressure—inches of water

and

$$W_2 = k\sqrt{P_c - P_f},$$

where

P_f = Furnace pressure—inches of water

Substituting for the constant, k,

$$W_2 = \frac{W_1}{\sqrt{P_c}} \sqrt{P_c - P_f},$$

or

$$W_2 = W_1 \sqrt{\frac{P_c - P_f}{P_c}} \text{ pounds per hour.}$$

The leakage ratio expressed in per cent is obtained from the equation

$$\text{Leakage ratio } (\%) = 100\left(\frac{\text{actual leakage } (W_2)}{\text{total air used—lbs. per hr.}}\right)$$

Air Leakage of Burner Registers.

The amount of air leakage through the closed burner registers of a boiler is usually too great to be measured with a 2 inch or 3 inch flow nozzle. Therefore, a large orifice is inserted in the uptake and air is supplied to the boiler casing air inlet from the Laboratory's blowers. A diagrammatic set-up is shown in Figure 7. The pressure drop through the burners is given by P_5 minus P_4. It is this value which is used as the abscissa when plotting

SET-UP FOR BURNER LEAKAGE

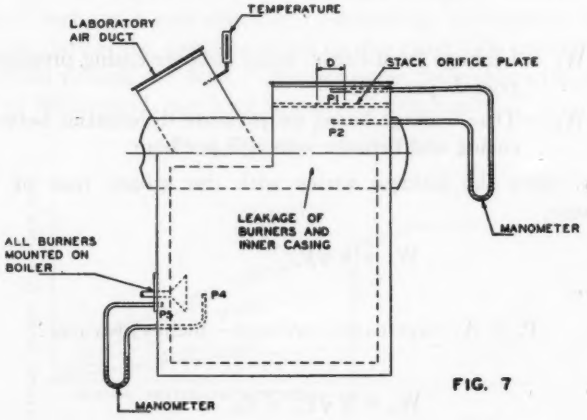


FIG. 7

the burner leakage curve. All air passing out through the orifice is the sum of the leakage through the burners and that through the inner casing because the outer casing leakage is negligible when the uptake is open as in this condition. The results are calculated by using the formula developed for the casing leakage, equation (4) above, d being the orifice diameter in inches. It is to be noted, however, that while the impact-flow nozzle as used in the casing leakage test has a discharge coefficient of nearly unity when using the subject orifice a correction must be made b multiplying the leakage determined from this formula by 0.6. Th pressure measurements are similar to those shown in Figure 1, with the pressure before the orifice, shown in Figure 7 as P_2, corresponding to the nozzle impact pressure and the pressure after the orifice corresponding to the nozzle static pressure. In order to find the leakage of burners alone, the inner casing leakage is subtracted graphically from the sum of the inner casing and burner leakage. The size of uptake orifice used for this determination varies with the number of burners on the boiler, one burner requiring an orifice of about 8 inch diameter. Therefore, for N burners the approximate diameter or orifice should be $8\sqrt{N}$ inches.

CIVILIAN TECHNICIANS, U.S.N.

By Lt. Commander W. T. Simmons, U.S.N.R.

The need arose early in the war to have experts available who were so completely familiar with the new special equipment that they could immediately diagnose trouble, assist in the establishment of bulk warehousing, identification of spare parts and the planning of spare parts depots, counsel the Commanding Officer to whom they were instructed to report, train Naval Personnel to become better "trouble shooters" on the equipment which they were operating, and analyze production methods and new scientific equipment developed by the enemy. These experts were designated as Civilian Technicians and were expected to conform to all rules and regulations promulgated in the interest of good order and discipline. In the presence of the enemy they were expected to conform with the actions of the Naval or Marine Corps personnel and not jeopardize the safety of the command. The status of a Technician was that of a noncombatant. The distinctive uniform and Certificate of Identification prescribed were intended to insure, insofar as practicable, that they would be treated as prisoners of war under the Geneva Convention in the event of their capture by the enemy.

Many types of civilian Technicians were required by the Navy and were made available by industries. The Bureau of Aeronautics, Bureau of Yards & Docks, Bureau of Ordnance, and Bureau of Ships utilized the services of these industrial representatives more extensively than other activities in the Navy Department.

This article will concern itself principally with those representatives who were sponsored by the Bureau of Ships. They include specialists in diesel and other marine engine maintenance, bulk warehousing of material, identification of parts for issuance and salvage, boilers, gyro-compasses, turbines, gears, purification of water, elevators for aircraft carriers, steering gears, cranes, winches, pumps, air compressors, governors, valves, power house switch boards, switch gear, voltage regulators, circuit breakers, minesweeping gear, propulsion control equipment, radio aids to air navigation, sonar and radar. Some of these specialists were

excellent mechanics, others were engineers, inventors, scientific and professional men at the top of their professions, all eager to help win this war.

When it was deemed advisable to send a Civilian Technician on board a Naval vessel or to a foreign station, approval from the Commanding Officer or Area Commander was received prior to flying the Technician to that location. The technical section in the Bureau of Ships communicated with the company whom it was thought could furnish the necessary men. A contract between the Navy and the organization was prepared. This compensated the organization for the services of their employees so utilized and compensated the Technicians for transporation, uniforms, passports, and innoculations if not supplied by the Navy. Approval was received from the Chief of Naval Operations, orders to duty were prepared, the Technician was briefed and given special instructions, and transportation arranged to his point of destination.

Let us suppose that he is directed to the Pacific Area. He arrives at San Francisco in the uniform of a Naval officer without rank, but wearing an embroidered spread eagle facing left with the left claw of the eagle clutching a group of tools, and the right claw an olive branch, while immediately underneath the eagle are the letters "U. S. TECHNICIAN". He is flown by Naval Air Transport Service or Army Transport Command to his destination. At that location he reports in, just as a Naval Officer would, and is assigned his duties. The regulations permit him, when possible, to mess and be billeted with Naval Officers. He performs the duties assigned, and at the completion of his work he is either forwarded to another location or is returned to the United States for further assignment. On arrival in the United States he reports to the Bureau of Ships.

These representatives from industries working with the Navy as Civilian Technicians, sponsored by the Bureau of Ships, performed meritorious services in all parts of the world. They were in the African, Italian, Philippine and Normandy invasions, aboard all types of vessels on the High Seas,—in the North and South Atlantic, the Caribbean, Mediterranean, Indian Ocean, Alaskan waters, Central Pacific, South Pacific, Southwest Pacific, —and in New Guinea, Guadalcanal, Borneo, Guam, Iwo Jima, Saipan, and Okinawa. They were in England, India, and Australia, setting up bases to keep the battle damage and spare

parts flowing to the vessels so that the Fleet could continue operations. They were among the experts who cleared up the intricate problems involving the operation and maintenance of radar and sonar equipment, helping to train the regular operators in the intricacies of new equipments as they were installed in the vessels. Technicians were in Germany and Japan observing the technical progress made by our enemies, and comparing their war equipment with our own. They helped in training of Naval personnel in the maintenance of amphibious equipment. They were "pinned down" on the beaches under fire. Some died in the performance of their duties.

NOTES.

INDEX TO NOTES.

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The Sural Boiler.
—Transactions of the Institute of Marine Engineers, Sept., 1945.

Design of Supercharged Diesel Engines.
—The British Motor Ship, Sept., 1945.

Grease Lubrication of Ball-Bearing Motors and Generators.
—Mechanical Engineering, Oct., 1945.

Shop Notes on Welding Stainless Steel.
—Welding Journal, August, 1945.

The Principles of Radar.
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A Marine Gas Turbine Plant.
—Proceedings American Society Naval Architects and Marine Engineers, Nov., 1945.

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—General Electric Review, Dec., 1945.

GERMAN CRUISER "PRINZ EUGEN".

This article, which describes briefly the machinery installations and arrangements on the 19,500 ton German Cruiser *Prinz Eugen,* is reprinted from the September 1945 issue of "The Marine Engineer".

On a visit to Wilhelmshaven at the beginning of July we were given facilities by Captain Conder, R.N., the Naval-Officer-in-Charge, to go over the German heavy cruiser *Prinz Eugen,* the most powerful warship of what remains of the German Navy. It was a somewhat strange experience being taken to the ship in a motor-boat manned by a German crew flying the White Ensign, the British party being piped aboard as it reached the upper deck of the warship. The vessel was lying in a remote part of the harbor, and made a fine spectacle in the sun with its towering bridge superstructure and twin-mounted 8-inch guns. She was manned by her German crew (most of whom were sunbathing at the time on sand dunes nearby) but flying a code flag, like other German warships with a German crew on board, prescribed by British Naval Headquarters.

The *Prinz Eugen* will be remembered by her sudden dash up-Channel in February, 1942, from Brest, where she and the *Scharnhorst* and *Gneisenau* had been under persistent air bombardment by British air forces, to take refuge in northern waters. She shared an inglorious existence with most of the other German surface vessels, lying idle for most of the war in port, with little attempt being made to employ her usefully in the service of her country.

If the conduct of the German war on the surface of the sea lacked inspiration and enterprise, the actual design and construction of this vessel reveals a wealth of ingenuity and inventive skill for which Germans have long been renowned. One is indeed puzzled during a single visit to the machinery spaces to understand how such a mass of intricate machinery was arranged in such a small space. We were informed that a complete mock-up of the propelling machinery was installed in wood prior to the fitting in of the actual machinery. It is difficult to obtain any clear idea of the machinery layout during a restricted visit owing to the cramped spaces for "walking round," much of the installation resembling a jig-saw puzzle on first acquaintance. From the point of view of maintenance the machinery arrangements, in our view, leave much to be desired, and in fact the vessel was in all probability designed with the idea that continuous steaming month in, month out, under wartime conditions, with few opportunities for engine repairs or overhauls, would not be required. Such conditions are, of course, the very opposite of what is required in a British warship, and it can be assumed that the general design would receive little favor with British naval engineers.

Having said which it is only right to state that, having decided on the type of installation, the Germans carried it out with skill and resource, and there are ideas (particularly in the region of indication and control of steam conditions at various points) which deserve examination. The illustrations which we reproduce convey a good idea of the way in which German engineers carried out their system of machinery operation and regulation from the various control positions in the ship.

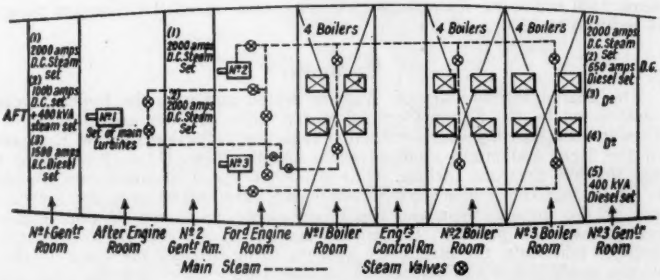
The *Prinz Eugen* was laid down in the Germania shipyard, Kiel, under the 1935 programme and was completed in 1940. She belonged to the *Hipper* three-ship class, the third vessel being the *Seydlitz* (ex-*Lützow*). She has a length of 654 feet 6 inch, beam of 71 feet and draught of 15 feet. Standard Displacement is 19,500 tons. Her vertical side armor has been stated to be 5 inches thick.

The vessel mounts eight 8 inch guns in twin turrets, 12 4.1-inch anti-aircraft guns, a number of quick firing guns, and has 12 triple-mounted torpedo tubes. She carries four catapult air-craft. She is propelled by

geared turbines driving on three shafts. Her flush upper deck runs all fore and aft, with a fairly steep sheer forward leading to a heavily raked stem. A short foremast (fitted at the after end of the bridge) and tall mainmast are fitted, and the boiler uptakes are brought to a single funnel surmounted by a characteristic well-sloped cowl. The ship's cruising speed is obtained on 80,000 Shp.

Machinery of 135,000 S.H.P.

The propelling machinery of the *Prinz Eugen* is undoubtedly one of the most advanced geared turbine installations afloat, comprising as it does three sets of high-powered lightweight geared turbines and an installation of forced-circulation water-tube boilers designed for very high pressure and temperature conditions. The total maximum power of the installation is 135,000 Shp., making the output per shaft one of the highest in a vessel of its type.



Disposition of the Five Main and Auxiliary Engine Rooms and Three Boiler Rooms of the "Prinz Eugen." The Engineers' Control Room is an Interesting Feature of the Vessel.

The general disposition of main turbines and boilers is shown on the accompanying sketch which gives an incidental reminder of the close subdivision which has been a feature of German warship design for some time past. For purposes of reproduction it should be explained that our rough layout has been compressed somewhat for in actual practice the engine rooms and boiler rooms extend over the greater part of the double bottom of the ship; and because of her fine form No. 1 generator room, the after engine room, No. 3 boiler room and No. 3 generator room are appreciably narrower in relation to their length than is shown on our plan.

Considerable thought has been given by the designers to ensuring that the vessel can still maintain a reasonable speed should any of the boiler rooms be seriously damaged. This is reflected in the steam piping layout and the disposition of the various steam valves, shown by means of circles on our drawing. It is normal practice in the cruiser to take steam from all three boiler rooms whatever the speed of the ship. Thus for all speeds up to 21/23 knots two boilers in each boiler room are normally in use, while from the latter speed up to 27 knots three of the four boilers in each boiler room are brought into service. From 27 knots up to the maximum of 32/33 knots all twelve boilers are employed. At the same time the arrangement of the steam piping and valves is such that a considerable number of combinations of boiler and steam arrangements can be employed so as to keep the machinery running in the event of damage making it impossible for the usual steaming practice to be followed.

Considerable importance was attached in the design stages to securing good cruising speed performance. To this end close attention has been given to providing a form which is easy to propel at the so-called economical speed, while the turbines are likewise considered to be very economical at the normal cruising speed of 15/17 knots; separate cruising turbines or diesels are not provided. The total fuel oil capacity is 3230 tons, although this can be increased to 4000 tons by using the bulge tanks. In addition 130 tons of diesel fuel oil can be carried, as well as 4½ tons of diesel lubricating oil. Some 116 tons of turbine lubricating oil is carried in the storage tanks with an additional 10 tons in each geared turbine installation. In addition to the foregoing 600 tons of feed water, 120 tons of drinking water and 160 tons of washing water are carried, making 880 tons in all. All this, with the fuel and lubricating oil figures given, represents a total of 5160 tons of liquid carried by the vessel. With the considerable quantity of fuel oil carried, coupled with fair fuel economy at economical cruising speeds, the endurance at 15/17 knots is some 5500 miles, we understand. At maximum speed the steaming range is reduced to 2500 miles.

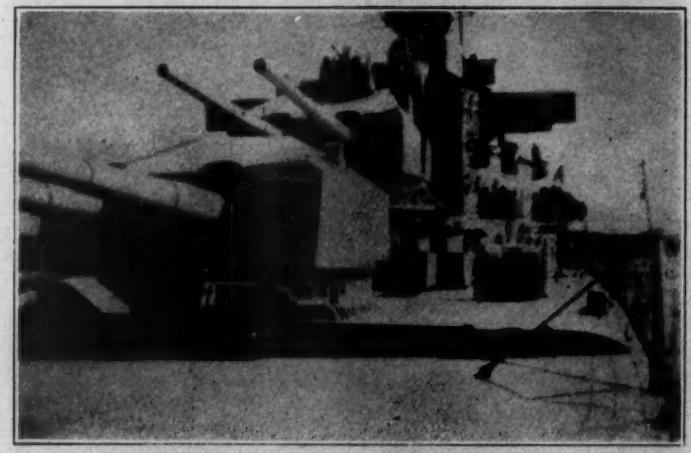
Machinery.

The geared turbines are of Wagner design and with the boilers, were constructed by the Krupp Germaniawerft. The turbines are of straightforward three-casing single-reduction type, the reduction ratio in the case of the high- and intermediate-pressure spindles being 12 to 1 and 9 to 1 for the low-pressure turbine. The propeller speed at maximum output is about 360 Rpm. The high-pressure and intermediate-pressure turbines are of the impulse type and the low-pressure units are of double-flow impulse-reaction pattern. In general design, layout and specification as regards materials, the machinery appears to be broadly the same as that of the modern German destroyers, although the turbine-gearing units are somewhat larger than those installed in the big *Narvik* class destroyers.

An interesting feature which we noted was that the reduction gearcases are of cast steel and not of welded steel construction as might be expected. In this connection it is well to recall that since before the last war the Germans have been skilled specialists in the production of complicated lightweight steel castings. Their submarine diesel engines of world war No. 1 furnished good examples of the stage to which they had advanced more than 30 years ago.

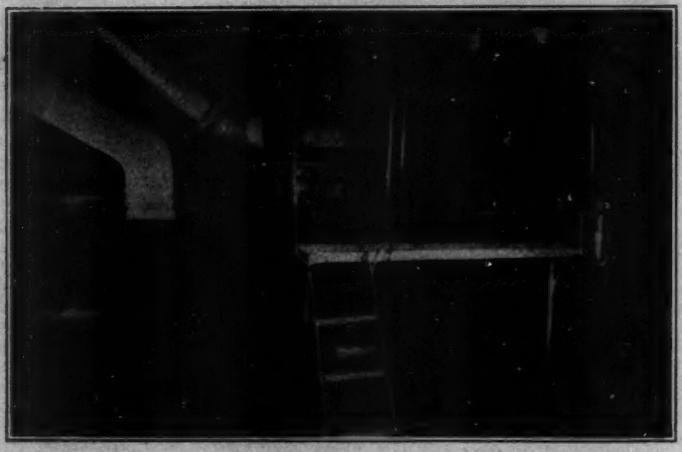
Each set of machinery has its own under-slung condenser, this being of the two-flow type. Each condenser has a vertical-spindle turbo-driven circulating pump of the high-speed lightweight type, this only being used when getting under way and when maneuvering. At approximately 10 knots in summer time the circulating pumps are cut out and scoop circulation is then relied upon up to maximum speed and power. In winter the circulating pump is only used for speeds up to 6 knots, above which figure a satisfactory vacuum can be maintained by scoop.

The closed feed system is employed, the extraction pumps being compact vertical-spindle turbine-driven units, as in fact are all pumps in the vessel connected with main propulsion and damage control. Particulars of the air ejectors are not known but it is recalled that the feed water leaves the first stage of the two-stage feed pump at a pressure of approximately 150 pound per square inch and is passed through a surface type feed heater where it is raised to a temperature of approximately 110 degrees C. (230 degrees F.), auxiliary steam being the heating medium. It then passes to the second stage of the feed pump and is delivered to the boiler economizer at a pressure at full output of, we were informed, 85 atmospheres or nearly 1250 pound per square inch.

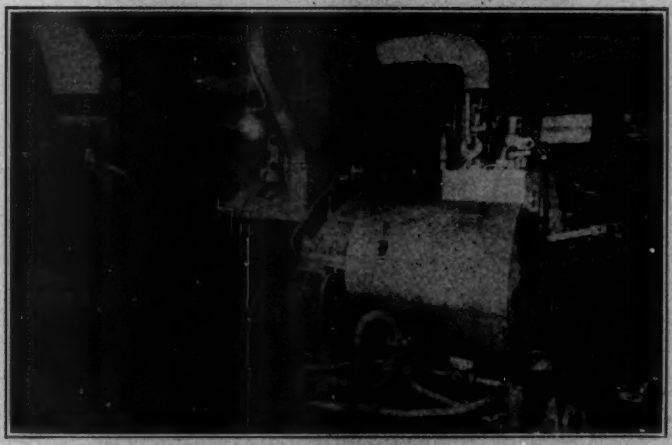


LOOKING FORWARD ON THE "PRINZ EUGEN".

Damage Control Section of the Engineers' Control Room.



The M.A.N.-Engined Direct-Current Diesel Generator in No. 1 Auxiliary Engine Room.



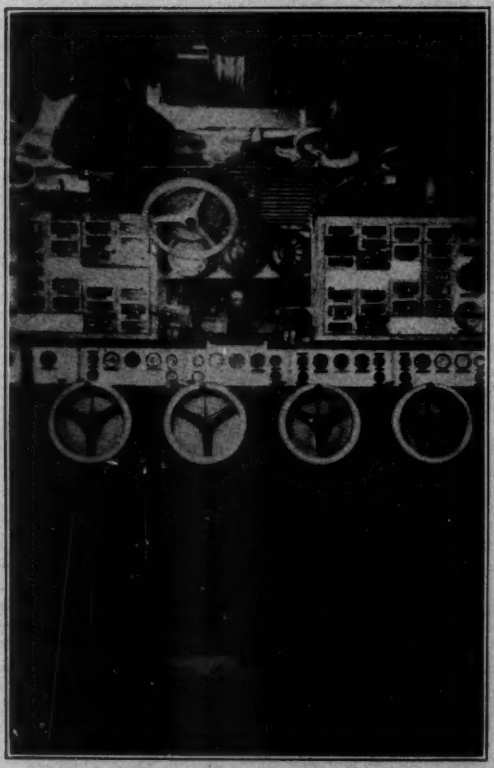
THE LARGE TURBO-GENERATOR IN NO. 1 AUXILIARY ENGINE ROOM.
THIS IS A GERMAN BROWN BOVERI MACHINE.

Looking Across One of the Boiler Rooms. Note the Vertical-Spindle Turbine-Driven Circulating Pump for the Boiler in the Foreground.

PART OF THE MAIN SWITCHBOARD IN NO. 1 GENERATOR ROOM.

PORT-SIDE OF THE MAIN CONTROL PLATFORM IN THE AFTER MAIN ENGINE ROOM.

COMPARISON SHOWS THAT THE IMPRESSIVE CONTROL STATION OF THE FORWARD MAIN ENGINE ROOM IS A "DOUBLE BANKED" REPLICA OF THE AFTER PLATFORM.



THE ASKANIA GEAR PANEL IN NO. 1 MAIN ENGINE ROOM. THIS EQUIPMENT AUTOMATICALLY CONTROLS COMBUSTION, STEAM TEMPERATURE, AIR PRESSURE, ETC.

The thrust of the solid bronze propellers is absorbed by single-collar thrust bearings of what would appear to be Michell type. The wing propellers are inward turning ones while the centre screw turns in the same sense as the port propeller. The shafts are carried on plain bearings. Westphalia centrifugal purifiers are provided for cleaning the turbine lubricating oil.

Boiler Plant.

The boiler equipment is particularly interesting for it must be the largest and most important La Mont marine boiler installation afloat today. Originally the twelve forced-circulation boilers were designed for a working pressure of 1130 pounds per square inch but this was later decreased to a figure of approximately 880 pounds per square inch, i.e., approximately the same as the pressure used in the German destroyer installation about which we have already written. The reason for this change was not too clear to us but we gathered that the trouble concerned the dynamics of the turbine-shafting system rather than the boilers. The total steam temperature at full power is 450/500 degrees C. (840/930 degrees F.). The boilers operate on the closed stokehold system of forced draught, with 14 inch water gauge pressure at full power. The forced draught fans are geared-turbine-driven, as are the compact vertical-spindle circulating pumps for the main boilers (two per boiler).

The boilers have very small tubes—about one inch in diameter—an interesting feature being the employment of Saacke rotary air-turbine-driven burners. There are only two of these burners per boiler, it is worth noting, the air supply to the burner turbines reaching them through the air jacketing around the boiler. The normal speed of the burners in this vessel is around 4000 Rpm., and from our examination they appear to be robust and of very nice construction. There is no doubt from observation and conversation regarding its behavior that the Saacke burner is efficient in practice, while it has the virtue of having an exceptionally high capacity. The normal oil fuel pressure is about 60 pound per square inch at the burners and the oil is heated to 45 degrees C. (112 degrees F.).

The oil burning equipment seems to have been the joint responsibility of the Germaniawerft and the Deutsche Werke and so far as could be ascertained presented no special features apart from the Saacke burners. When starting up the boilers motor-driven forced draught fans are used and once sufficient steam is available the turbine-driven fans take over and the electrically-driven ones are shut down. Steam can be raised to normal working pressure in 45 minutes if the boilers are initially empty, and in less than two hours if they are full to normal working level.

In general design the boilers appear to be normal La Mont drum-type forced circulation units, the design incorporating superheater, plain tube economizer and tubular air heater. The steam pipes throughout the vessel have screwed flanges and expansion is taken care of by corrugated sections wherever necessary. Auxiliary steam from the main boilers is supplied at 150 pounds per square inch for some purposes and at 78 pounds per square inch for other services, this being drawn from the boiler before the superheater and passed through suitable reducing valves. All cooking throughout the vessel is done by steam and heating of accommodation, etc., is attended to by steam at 150 pounds per square inch pressure.

When in harbor two geared turbo generator sets can carry the electrical load, their working pressure being 370 pounds per square inch. These units employ superheated steam but those auxiliaries which operate at 150 pounds per square inch use saturated steam. As was mentioned in connection with one of our destroyer descriptions the evaporating equipment is arranged to work on a closed circuit and its capacity is appreciably in excess of actual make-up feed requirements. At normal full power some 150 tons of make-up feed is required per day by all boilers whereas

the six evaporators are each capable of producing 50 tons of fresh water per day. When individual evaporator output drops to 35/40 tons per day they are descaled, this being necessary about every 800 hours in the North Sea. In the fresher waters of the Baltic, however, the evaporators can usually be run for 1000 hours before needing an opening up.

Like the German destroyers the *Prinz Eugen* is provided with Askania automatic gear for controlling main machinery, boilers, etc. Each boiler has its own control unit for adjusting air, fuel oil and feed for optimum conditions at any required output, while automatic control of the various steam-driven auxiliaries and motor driven fans in the machinery spaces is likewise attended to by the Askania gear.

The accompanying plan showing the machinery layout gives an indication of the number, size and disposition of the various generating sets. It remains to be added that the geared turbo generators are of German Brown-Boveri make, while the diesel engines are six-cylinder M.A.N. units of medium-high-speed type. Both turbine- and diesel-driven generating sets are illustrated with this article.

Unfortunately we were not able to see the machinery of the *Prinz Eugen* running but the general impression which the installation left after a very thorough inspection of all the engine rooms and boiler rooms was that the design is exceptionally up-to-date, while good use has been made of the space. To British marine engineers the installation is undoubtedly complicated, and from the information given regarding hourly fuel rates this seems to indicate that the specific fuel consumption at full power is not exceptionally good. We were not given the power at the so-called economical speed and so cannot offer an opinion on the performance at 15/17 knots. The *Elbing* class torpedo boat with equally modern machinery was not exceptional at partial loads, as we have indicated and probably the same applies to the *Prinz Eugen.* As an example of German naval construction and marine engineering she is, however, a noteworthy vessel and will provide Allied naval engineers with considerable food for thought and discussion.

THE SURAL BOILER.

This translation of an article in "Journal de la Marine Marchande" of December 1944 by J. P. Ricard as it appeared in the "Transactions of the Institute of Marine Engineers" of September 1945 outlines the principles of operation of the French Sural forced combustion type boiler.

If instead of carrying out the process of combustion in the furnance at a pressure slightly above that of the atmosphere, the combustion pressure is deliberately raised to something over 28 pounds per square inch, the velocity of the gases is considerably increased. With a pressure of the order of 30 pounds per square inch abs. the gas velocity is between 7000 and 8000 feet per second. This velocity, added to that of the water-steam emulsion in the tubes, permits a very high intensity of heat exchange to be effected, an intensity which, in fact, makes it possible to produce 75000 pounds of steam per hour with 668 feet of heating surface, whereas in the case of a cylindrical boiler, this evaporative capacity would require over 23000 square feet of heating surface, *i.e.,* about 35 times as much.

The main advantages claimed for forced combustion are the reduced amount of weight and space required for the generation of a given amount of steam, but a further important benefit which has, in the opinion of the writer, not yet received the attention it merits, is the *improvement of the actual process of combustion.* It is suggested that this is a matter which calls for systematic research work for the sake of the improvements in

the design and construction of merchant-ship boilers which would result from it.

In theory, the process of combustion is not dependent on the pressure under which it is carried out, but experience with Diesel engines has, nevertheless, shown that the secondary and mechanical factors of combustion are bound up with the latter; these factors are the atomization of the fuel, the intensity of the mixture of air and fuel, and the turbulence of the combustion. Even if past operating results have not pointed to the fact that there is still something to be learned in regard to these matters, it is clear that an increased velocity of the gases must result in a greater elimination of unconsumed particles of fuel, cinders and other impurities which have, during the past few years, been responsible for much of the trouble given by modern watertube boilers when burning oil fuel of the relatively poor quality which one has frequently been obliged to utilize in merchant vessels.

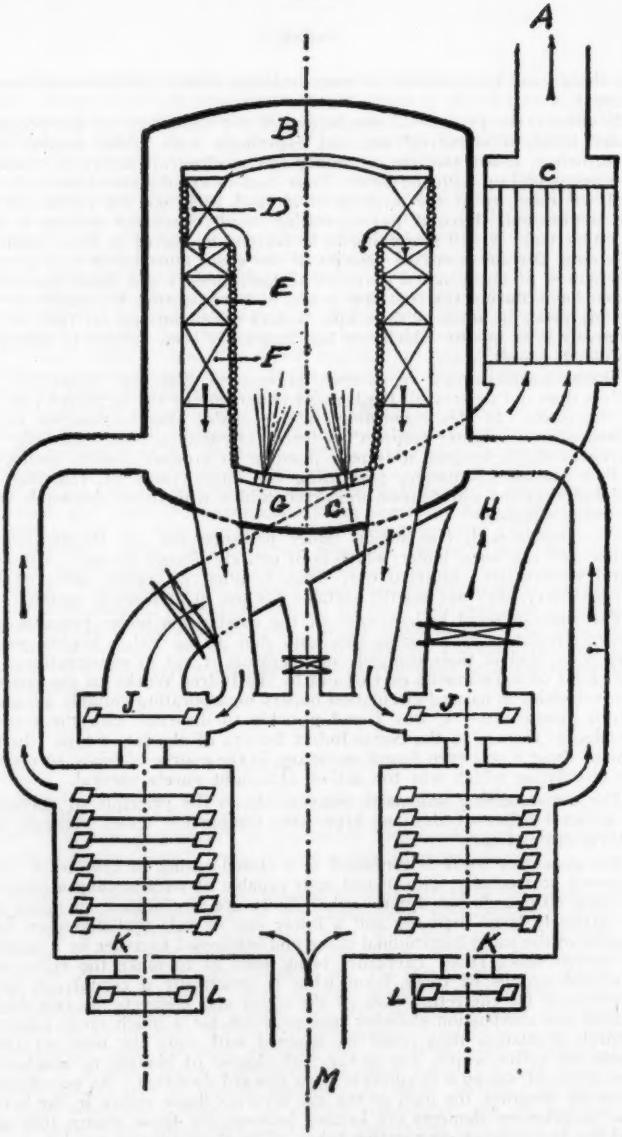
These considerations, of course, presuppose that the design of the boilers does not destroy all the benefits obtainable by the increased velocity of the gases. In this connection, it is probable that combustion under pressure has a brighter future than forced circulation, more especially for merchant ships, because it is now possible to produce boilers with controlled natural circulation, permitting the employment of velocities of circulation of the water-steam fluid comparable with those obtainable with forced circulation.

The boilers with combustion under pressure are the Brown Boveri Velox, and the Sural boiler, which is of entirely French design. Although Sural boilers have hitherto only been installed in certain units of the French Navy, they are equally suitable for use in mercantile vessels.

The earliest Sural boilers were of the combustion under pressure and forced-circulation type comparable with that of the Velox boilers, which they resembled in their principle of operation if not in construction. As the result of experiments carried out by the Indret Works on the circulation velocities in natural-circulation boilers incorporating suitably arranged return downcomers, it was found possible to dispense with the forced-circulation feature in the Sural-Indret boilers of the latest type. In the case of these it was even found necessary, in the course of trials, to restrict the circulation which was too active, althought purely natural.

The accompanying schematic diagram shows the principle of operation of a Sural boiler of this last type with combustion under pressure and natural circulation.

The generator itself is contained in a closed casing of cylindrical form disposed horizontally, with dished ends capable of withstanding a pressure of some 40 pounds per square inch. There are two drums, an upper one of relatively large diameter and a lower one of half that diameter, both located in the same longitudinal plane and connected together by a number of curved tubes (their curvature being such as to make the right- and left-hand groups of tubes form what is practically a circle). A large downcomer for the return flow of the water also connects the two drums outside the combustion chamber and provides for a much more intensive natural circulation than could be attained with only the nests of tubes inside the boiler which, due to the high degree of heating to which they are subjected, set up a circulation in an upward direction. As may be seen from the diagram, the path of the gas involves flame return in the boiler. The superheater elements are located between the flame return tube nest and the last nest of evaporative tubes. The diameters of the tubes vary from 25/30 mm. (=1.0/1.2 inch) to 40/48 mm. (=1.6/1.9 inch), according to the nests and their location. In general, the Sural boiler does not differ from the usual types of watertube boiler, except for its cylindrical form. As in those boilers, the gases circulate around the tubes, whereas



PATH OF FLUE GASES IN SURAL BOILER.

A, uptake; B, closed casing; C, economizer; D, flame return tube nests; E, superheater; F, evaporative elements; GG, oil fired burners; H, gas outlet; JJ, gas turbines; KK, turbo-blowers; LL, steam turbines; M, air suction of blowers.

in the Velox boiler the whole conception of the tube system is radically different.

Like the Velox, the Sural boiler is equipped with an economizer heated by the exhaust gases from the gas turbine driving the air blower. It should be noted, however, that the method of air pre-heating used in normal type watertube boilers is not applicable to combustion under pressure boilers, since the compression of the combustion air in the turbo-blower is sufficient to raise the temperature of this air to between 250 degrees and 280 degrees F. under ordinary operating conditions.

Every Sural boiler is equipped with two turbo-blowers, each comprising a multi-cellular axial compressor driven from one side by a gas turbine and from the other by a steam turbine. The former is able to provide all the power required to drive the blower when the boiler is being steamed at a high rate; it is, however, unable to develop this power when the boiler is being operated under light load, and the steam turbine is employed to drive the blower from the time of lighting the O.F. burners until the boiler is generating its full output of steam.

The trials carried out with these Sural boilers, which are far simpler than generators with combustion under pressure and forced circulation, are reported to have proved highly satisfactory. An interesting feature of the Sural boiler is that it is not equipped with soot blowers, the high velocity of the flue gases sufficing to keep it clear of soot and other solid by-products of combustion.

DESIGN OF SUPERCHARGED DIESEL ENGINES.

This is a reprint of the extracts from a paper presented by Mr. R. L. Boyer, Chief Engineer of the Cooper Bessemer Corporation, before the Oil and Gas Power Division of the American Society of Mechanical Engineers as they appeared in the September 1945 issue of "The British Motor Ship".

It has become a popular belief that all one has to do in order to increase the output of a given engine by perhaps 50 per cent is to add a supercharger.* While supercharging is an ideal way of increasing the output of an engine and does produce considerably more horsepower for a given weight and given spare requirement, it is, unfortunately, not as simple a matter as just indicated; and there are certain design considerations which must be taken into account. The purpose of this paper, therefore, is not to discuss details of supercharging, but to point out some of the experience gained from the engine-design standpoint.

The information contained in this paper has been acquired through experience with engines from 8-inch bore to 15½-inch bore and with speeds from 277 to 1000 Rpm. No attempt has been made to cover the experience of others.

The earliest application of supercharging was without change of valve design, and was therefore to engines having two valves, or one inlet, and one exhaust. Many successful supercharging applications have been made on two-valve engines. However, due to the absolute necessity of thorough scavenging, it was soon discovered that the four-valve engine, having two inlet and two exhaust valves, was much more thoroughly scavenged and was, therefore, much more adaptable to effective supercharging. For that reason, all later designs have been based on the four-valve type, and all supercharged engines now manufactured by the author's company are of the four-valve design.

* The author uses the term "supercharging" as synonymous with "turbocharging," since the turbocharged engine is the type with which he is most familiar.

While unquestionably the four-valve design is somewhat more costly, it is believed to be well worth the additional cost, considering the more effective supercharging and, therefore, the higher ratings obtainable. Another reason for the four-valve construction is that, since the piston head must contain pockets to clear the valves, a four-valve engine design produces a piston head which is more symmetrical and which is more likely to produce clean combustion. This will be discussed later in more detail.

As noted, scavenging is an extremely important phase of effective supercharging. It is believed that the thorough scavening resulting from a supercharging system can be responsible for as much increase in output as the supercharging itself. In order to get this effective scavenging, it becomes necessary to have a materially increased overlap of timing of the inlet and exhaust valves. In other words, the exhaust valve is held open longer and the inlet valve is opened earlier in order to provide sufficient overlap to insure a considerable amount of air going directly from the inlet through the exhaust valve, thereby providing a thorough removal of residual gas. Comparative timing diagram, showing the resulting overlap in the case of supercharged engines, are illustrated in Figure 1. Considerably less overlap is usually employed where mechanical blowers are used. As indicated, however, the purpose of this paper is to present purely the experiences with turbocharging.

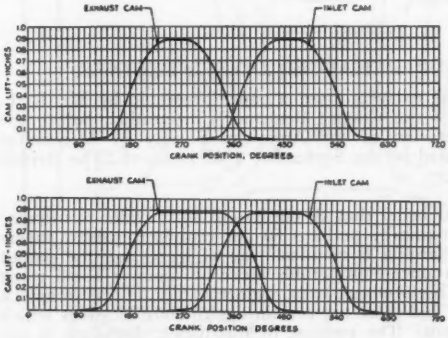


Fig. 1 (left).—Upper and lower respectively—cam-lift curves for atmospheric and supercharged Diesel engines.

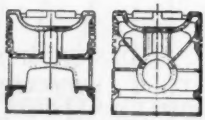


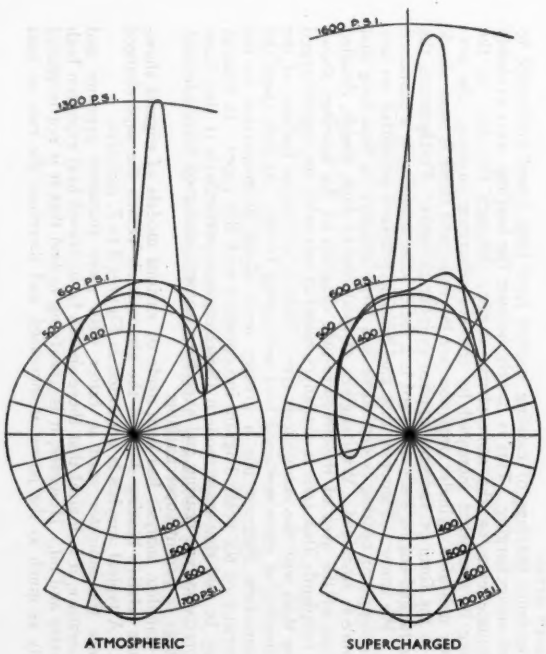
Fig. 2 (above).—Example of efficiently cooled piston.

In order to get the altered timing diagram it is, of course, first of all necessary to provide entirely different cams. This, therefore, becomes the first "must" when considering design alterations.

Pistons.

Reference has already been made to the fact that pockets must be provided in the piston heads to clear the valves. Diesel combustion chambers are so shallow that, with direct-injection engines, pistons must usually be notched at the top to clear the valves, even in the atmospheric engine. With the supercharged engine, since the valves are held open much longer and are open an appreciable amount when the piston is on top dead center, these pockets become quite deep. This usually means not only machining the pocket, but also a change of the core contour of the piston in order to permit this machining. Therefore, to supercharge an engine effectively a different piston is usually required.

Since the cylinder pressure is maintained over a longer period of time with supercharging, there is naturally more heat dissipated from the com-



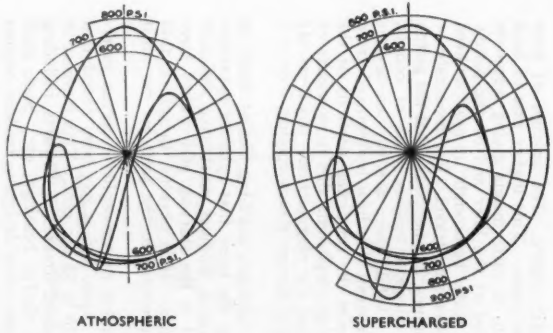
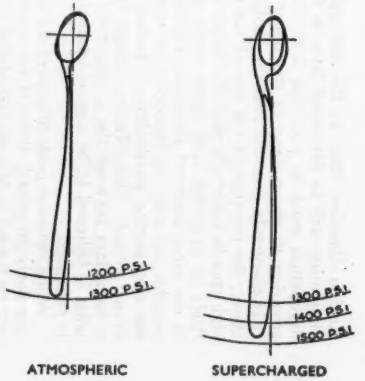


Fig. 3 (left).—Crankpin bearing pressures.

Fig. 4 (above). — End main bearing pressures; Nos. 1 and 9.

Fig. 5 (right). — Main bearing pressures; Nos. 2, 4, 6 and 8.



bustion chamber than in the atmospheric engine. The amount of heat dissipation will be discussed later in this paper. This higher rate of heat dissipation must be provided for, which, in turn, means giving adequate attention to the cooling of the upper end of the liner and of the piston. It is absolutely essential that the cooling water be brought up near, or even above, the position of the top of the piston-ring travel. For moderate or high speeds, especially with the larger engines, it is necessary that the piston itself be provided with some form of adequate cooling. Oil, of course, is the most accepted means of cooling. While there are various designs possible, all must be based on the requirement that heat be removed from the ring section particularly, whether from the remaining portions of the crown or not. One effective design for accomplishing this is by means of a heat dam between the crown itself and the ring section. To illustrate what is believed to be a good example of efficient piston cooling, Figure 2 is presented.

While cylinder peak pressures do not increase anything like as much as popularly supposed, there is no evading the issue that cylinder pressures are increased in the supercharging process. Engines must have sufficient compression pressure to start cold, and in the starting period they have little if any effect from the supercharging. It is obvious, therefore, that when we now add the supercharging pressure to the inlet air we must expect an increase in the peak pressure.

A study has been made of a large number of atmospheric versus supercharged engines of otherwise identical types, and it has been found that the cylinder pressure usually increases 10 to 15 per cent. This increase in pressure naturally reflects itself in increased stresses of all parts, including pistons, connecting rods, main frames, shafts, bearings, etc. Such peak pressures are, of course, taken into account when designing modern engines which are to be supercharged. Even if converting an atmospheric engine to supercharging, assuming it to have been designed originally with reasonable conservatism, an increase of 15 per cent in cylinder pressure it not likely to be of any serious consequence in reflecting itself in increased stress.

Sometimes bearings of existing engines have been found inadequate to permit supercharging. This is largely because the bearings were probably inadequate even on the atmospheric engine, although the duty on the bearing is somewhat increased by the supercharging process. As previously indicated, cylinder pressures are increased and there is more shock load in the form of suddenly applied cylinder pressure. Furthermore, the supercharged-engine indicator card is a fatter one, resulting in a sustained load. We know, of course, that the bearing pressure permitted on any given design of bearing is dependent in a large measure on how long that pressure is sustained. With engines of moderate or high speeds, detailed analysis of the area of the pressure-time diagram of the bearings shows that frequently this area is actually somewhat less with the supercharged engine than with the atmospheric type. This is, of course, because of the combination of more sustained load and inertia forces. At zero speed the pressure-time diagram has a greater area, about in proportion to the appearance of the polar diagram incorporated with this paper. In general, it may be said that most modern Diesel engines, particularly if they have the precision-type bearing and have been properly designed, have adequate bearings to permit supercharging.

A complete analysis has been made of various models of engines showing the relative bearing duty of the atmospheric versus the supercharged type. A typical anaylsis is presented in Figures 3 to 7, inclusive.

Since the indicator card is fatter with more sustained pressure and temperature, it is logical that there should be increased heat rejection both to water and oil. However, it is gratifying to find that it is not increased nearly as much as the power is increased, and therefore the rate of heat

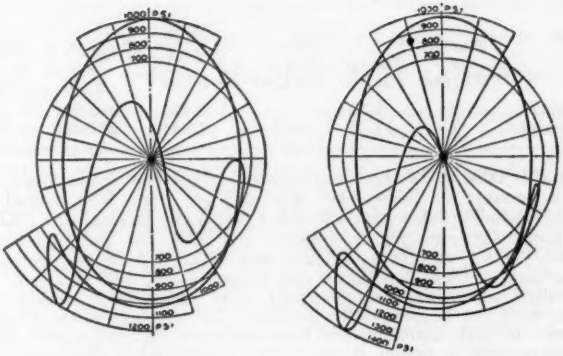


Fig. 6.—Main bearing pressures; Nos. 3 and 7.

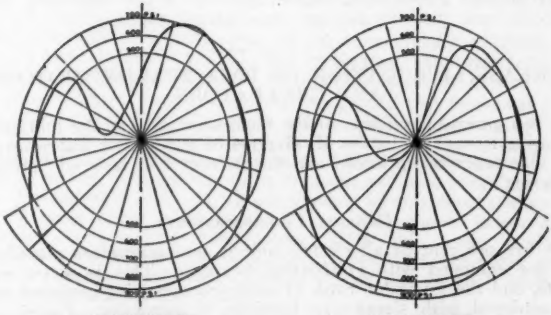


Fig. 7.—Centre main bearing pressures; No. 5.

rejection is materially reduced. To make a detailed analysis of this, a study of 11 engines was undertaken, five atmospheric and six supercharged. This analysis is presented in Table 1.

TABLE 1.—RATE OF HEAT REJECTION AND HORSEPOWER INCREASE OF ATMOSPHERIC AND SUPERCHARGED DIESEL ENGINES.

	Number of Engines	*Average b.m.e.p.*	*Increase in h.p. Per Cent*	*Average B.t.u. per h.p. per hr. to Water*	*B.m.e.p. x. B.t.u. = K*	*Increase in B.t.u. Per Cent*
Atmospheric...	5	82.6	....	2,482	205,000	....
Supercharged..	6	117.7	42.5	1,841	217,000	5.85

From this table it will be seen that the average increase in brake mean effective pressure or horsepower was 42.5 per cent. The overall heat rejection, including lubricating-oil cooler and water jackets, was 2482 Btu. per Hp.-hr. for atmospheric engines, and 1841 for the supercharged engines. Table 1, therefore, indicates that with a horsepower increase of 42.5 per cent the heat dissipation was increased only 5.85 per cent. This means that heat-dissipation equipment, with radiators or jacket water coolers, need not be materially increased for normal ratings of supercharging. In fact, assuming the heat-dissipation equipment was reasonably conservative in design, it should be adequate when changing over to supercharging. Water pumps present a similar situation, since the rate of flow need not be increased appreciably.

It is difficult to state properly the heat dissipation to lubricating oil in the atmospheric versus supercharged engines, because of variations in piston design. Data collected from a large number of engines indicate that the heat flow to the piston is increased about in proportion to the general heat dissipation indicated in Table 1, although, in some cases, it is somewhat greater.

A study of a large number of engines going over the test floor reveals that the average atmospheric engine consumes 5 per cent more fuel per horsepower-hour than the average supercharged engine.

GREASE LUBRICATION OF BALL-BEARING MOTORS AND GENERATORS.

This is a general discussion of the factors involved in the lubrication of ball-bearing motors by Edwin M. Higgins of the Master Lubricants Company. The paper is reprinted from the October 1945 issue of "Mechanical Engineering".

GENERAL CONSIDERATIONS.

Sixty-five per cent of all electric motors and generators manufactured today are equipped with antifriction bearings. This is a conservative estimate, and the favorable ratio of antifriction-bearing-equipped units to those equipped with sleeve-type bearings is increasing. Based on this estimate, it is evident that the lubrication of antifriction bearings is of universal interest. It is the purpose of this paper therefore to submit for consideration the general concepts of grease lubrication as applied to electric motors and generators although in many cases these concepts may be applied to other applications of antifriction bearings as well.

Why Grease Lubrication? The question is often raised as to the advantages and disadvantages of grease versus oil lubrication for antifriction bearings. It is recognized that each has advantages and dis-

advantages. Grease lubrication permits simplified bearing and housing design, ease of lubrication retention, versatility of mounting position, and, last but not least, the ease of maintenance. These have resulted in lower-cost equipment, cheaper installation, and more economical maintenance. This has caused grease lubrication to be used in all but the most unusual applications, and lubrication research is mainly directed toward the replacement of oil by grease in all applications.

Causes of Bearing Failures. The causes of bearing failures may in general be grouped as follows:

1. Poor bearings, that is, design, workmanship, and materials of the bearings themselves.
2. Faulty equipment caused by poor design, workmanship, or materials of the equipment to which the bearings are applied.
3. Misapplication of bearings, apparatus, or lubrication.
4. Bearing overloads caused either by careless operation of the equipment or the failure of some part of the apparatus system.
5. Lubrication failure or bearing contamination, in both cases caused either by faulty lubricating materials, careless maintenance, or overhaul.

Since lubrication cannot correct conditions which exist as a result of causes 1, 2, and 4, only causes 3 and 5 will be considered in this paper. Cause 3 will be dealt with only where misapplication of lubricants is concerned. Cause 5 will be considered in the light of maintenance and overhaul and, to a certain extent, trouble prevention by inspection of lubricating materials.

Selection of Grease.

In general a grease should be selected with the following taken into consideration:

1. Size and type of bearing and housing construction.
2. Speed of operation.
3. Ambient operating temperatures, or temperature range.

Greases may be typed in many ways, namely, type of soap base, viscosity of oil, relative consistency, quantity ratio between oil and soap base, melting point or drop point, and so on through the entire list of chemical and physical properties of the grease.

It might be well to discuss some of the commercially available types relative to their chemical and physical properties since many of these properties play an important part in the application of the grease.

Lime-Base Greases. Long experience has proved that lime-soap greases are not suitable for general-purpose use in ball-bearing motors. Their melting points are too low especially for 40 to 55 degrees C. temperature rise rated equipment. Lime-soap greases separate badly after being in use a short while; greasing has to be frequent and as a result the motors become contaminated with oil. Most greases of this type contain from 2 to 5 per cent water which remains from the original larger amount incorporated during their manufacture. Once this water is lost by evaporation, constant shear, or by higher temperatures, the soaps and oils separate, leaving soap residues. This is detrimental to good lubrication, and bearing failure is a likely result.

It might be well to mention here that oil seepage is highly undesirable, not only from its possible destructive effect on windings, insulation, and commutators, but dirt easily adheres to it, gumming up operating parts.

It is possible to produce anhydrous calcium grease which does not have such a tendency to separate, but very little is generally known as to the value of this type of material.

Aluminum-Base Greases. Most aluminum-base greases are almost transparent and are commonly called solidified oils. While they do not show the rapid separation rate of lime-base soaps, they can be classed

very closely to them as undesirable. Their melting points are low and with elevated temperatures they become more and more viscous or stringy. This increases fluid-friction rates, and the grease finally becomes either rubbery or is converted to a chalklike mass.

Soda- or Soda-Lime-Base Greases. Practically all specifications covering general-purpose ball-bearing greases call either for soda- or soda-lime-base material; these include the specifications as outlined by all ball-bearing manufacturers. However, just because they are that type is not the answer. We all know there are inefficient as well as high-grade greases in these classes.

Long-fiber or sponge greases are not desirable. Most of them show too much separation in service. They stick to and ride the retainers of the bearings, aggravating by centrifugal force this tendency toward faster separation. Solids or near solids rapidly form, gradually choking ball or roller action. Electric-motor lubrication presents a far different problem than the front wheels of an automobile.

Greases for antifriction bearings should not be stringy or tacky; these types develop undesirable friction within themselves. They should either be smooth in texture or very slightly fibrous.

A highly important characteristic is that the grease should have a slight tendency to separate so that the races and balls are a little "wet" and not dry in appearance. This is true lubrication. The exact extent of desirable separation is controversial but bearing failures leave little room for doubt that extremes in either direction are detrimental.

If the grease pack remains "dry" and a space finally develops between the grease pack and the bearing, a time arrives when the bearing itself needs more lubrication. In this case the only way the grease will move is by a temperature rise. All greases of this type have high melting points, and a temperature rise too great for the safety of the bearing would have to occur in order to move any grease into the bearing.

Lithium Greases. While lithium-base greases have high melting points, higher than most soda soaps, this feature cannot be used as a yardstick in measuring their ability to operate successfully in high temperatures combined with high speeds. In fact, research data at this writing is so inconsistent it would seem that much further investigation should be accomplished to classify this type of material properly. It is for this reason that careful consideration be given to operating conditions before a lithium-base grease is selected.

It is recognized that greases of this type have proved highly successful for operation in extremely low temperature or where water or moisture resistance is a desirable feature.

Inverted-Soda-Soap Greases. Inverted-soda-soap greases in general have the same characteristics as ordinary soda-soap greases and in addition they have higher melting points. They are water- and moisture-resistant if no more than a temperature of 120 F. is reached. The principal disadvantages of this type of grease are the extremely low separation rate even under elevated temperatures and the tendency to harden under high speeds.

As a result of these conditions, this type of grease does not operate well at elevated temperatures and speeds and makes their usefulness in this field questionable.

Recent research, even though inconsistent, has shown that this type of grease warrants further development.

Methods of Evaluating Greases.

Up to the present time no standard test procedure has been formulated for evaluating greases, with the exception of the approved A.S.T.M. physical tests for soap base and percentage, oil viscosity, penetrations, acid-alkali determinations, contamination, fillers, and moisture.

It must be admitted that even today grease making is not yet an exact science, it is still to a large degree a grease-makers' art which cannot be written into a physical test. It is for this reason that physical tests must be considered only primary. An unknown number of greases can meet the usual specifications laid down, including some government requirements, but too few perform satisfactorily.

Much effort has been made in the past few years to develop performance-test equipment, but as yet none of these machines has been accorded universal approval. Each machine is developed more or less to cover the requirements of the particular equipment of a given concern and is of little value when applied to other equipment.

The Annular Bearing Engineers' Committee has set up some standards of its own using the oxygen bomb and the A.B.E.C. testing machine.

The oxygen bomb has real value in that it can closely predict stability or storage life of grease in bearings. It has value in choosing lubricants for high-temperature application. It has little or no value in establishing operating characteristics.

The A.B.E.C. testing machine has the ability to eliminate unsatisfactory greases quickly, but the time element in making these tests is so short that its performance prediction for long service periods which are normally required in the field may fall short in accuracy. It does not give the better lubricants a chance to be graded properly. Even greases which have not performed so well in this test have shown very satisfactory service records in the field.

Much interest has been taken in the pressure viscometer and entirely satisfactory bleeding-test equipment has been built.

Cold-test equipment has been built which has shown real value.

It is difficult to incorporate even a few of the many variables which the many types of equipment offer and combine them in one testing machine. Building test equipment for the observation of reactions to one variable, another unit for something else and trying to put them together on paper does not work too well. Accelerated tests can prove very misleading.

An A.S.T.M. Committee is now being organized to work out this entire problem, and it is hoped that this committee will eventually produce equipment that can be used with more universally consistent results.

At the present time the best way to determine the performance of a lubricant for a given assignment is to put the test greases in units to be operated under natural conditions and give them a year's run in actual or simulated service, or for any length of time, short or long, for which satisfactory service should be shown. It is well not to make decisions too rapidly. If a decision must be made without delay there is nothing to be done but be guided by the limits of experience.

Consistency of General-Purpose Grease. There are varied opinions on this subject, but experience indicates that a range of from 250 to 300 A.S.T.M. worked penetration generally operates very well. If the grease is much softer it might churn and pump out along the shaft as well as cause higher than necessary operating temperatures. If it is much stiffer continuous supply might be interrupted, and without more frequent lubrication bearing failure is likely to result.

Heavy greases also offer mechanical-handling difficulties.

Very little work has been accomplished on determining penetration values over wide temperature ranges, especially the correlation of curves obtained with actual performance. Recent investigation of greases submitted for high-temperature high-speed operation gave the following trends; there being an interesting correlation between these curves and lubrication characteristics:

Grease No. 1 in Figure 1 shows high susceptibility to temperatures in all ranges and it was not considered as useful material.

Grease No. 2 has an interesting temperature-range curve, but the fact that it is so soft brought up the question of its ability to remain in the housing.

Grease No. 3 shows undesirable hardening characteristics after a temperature of 100 C. is reached.

Grease No. 4 has a consistent curve that proved satisfactory in high-temperature tests combined with high speeds.

Grease No. 5 would appear to be too hard at any workable temperature.

New grease might churn for a short period until it finds its steady working position as a grease pack.

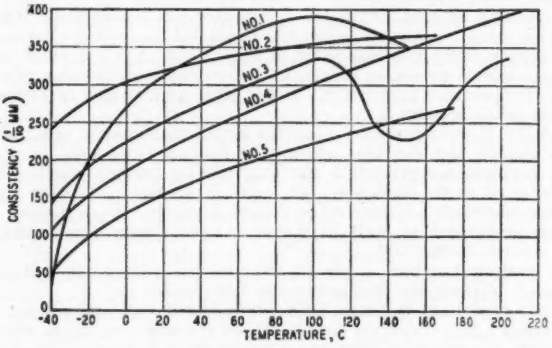


FIGURE 1.—CONSISTENCY VERSUS TEMPERATURE OF BEARING GREASES.

At normal operating temperatures, heavy greases are liable to offer only borderline lubrication or provide a film so heavy that it will shear at points of rubbing contact. This is particularly true in all types of roller bearings.

Color of Grease. Color itself of course has nothing to do with the lubricating ability of the grease. It usually is controlled by the color of the oil used in its manufacture. Some greases are dyed for reasons of identification, trade purposes, or color uniformity, and in some instances to disguise a product of poor appearance.

It is not always easy to keep grease clean and since contamination is one of the things we fear, a light-colored grease is preferable to a dark product of equivalent value as contamination can be more easily seen and remedied.

Dark-colored greases do not lend themselves to a quick visual determination of condition changes which usually mark the progress of oxidation. If a definite system of cleaning bearings is not in effect, the color change of a grease serves as an indicator and old grease can be washed out and renewed at the proper time.

If the pressure-relief system is used for greasing, the difference in the color of the new grease and the old makes it easier to determine when the new grease is coming from the drain hole. It may be seen again that a light-colored grease is preferable to a dark grease of equivalent properties.

HOUSING DESIGN AND ITS RELATION TO GREASING.

We constantly read and are told that an antifriction-bearing housing should contain no more than a one-third to a one-half filling of grease. It is true that this is an ideal quantity but this advice is not followed by a

suggested method by which the amount can be maintained over a period of years with periodic greasing. Original fillings can be quantity-controlled, but from then on guesswork must be the rule. Quantitative applications have been successfully arranged for railroad traction-motor equipment to the point of "ounces" added twice a year. This has been accomplished after much study, and it works.

The general factory problem presents greater complications, keeping in mind that there may be several hundred or several thousand different sizes and makes of motors in use with several men applying the lubricants. This group constantly changes in personnel, and no quantitative greasing records are maintained.

The advice therefore to keep housings one third to one half full is highly idealistic, and its steady accomplishment practically impossible.

The over-all picture of motor design prompts the author to offer the suggestion that there is still much room for the study and development of better designed housings so that good accurate lubrication systems can be provided in the interests of everyone concerned. Progress in this direction will greatly help in lowering maintenance costs.

While streamlining has made machine tools and other equipment more pleasing to the eye, it has also many times made the application of lubricants to motors more difficult. Machine designers and plant engineers should give more consideration to the maintenance problem; the ease of approach to both sides of motors without undesirable structural interference, wherever possible.

At present there are several housing designs and recommended means of lubricating motors.

Some have no drain openings at all, some are equipped with offset openings. The best designs have drain openings at the bottom where they are easily accessible.

In the first group the only way to grease is to start off with a one-third filling, adding a small amount of grease every 6 months. It is impossible to clean these housings without disassembly.

In the second group it is possible to use the pressure-relief greasing system but it is not possible to clean without disassembly as a quantity of cleaning oil and solvent would remain in the housing.

The last group offers the best conditions for pressure-relief greasing systems and cleaning without disassembly. The arrangement provides time, labor, production, and bearing-abuse savings beyond any other design or mechanical arrangement in the interests of long bearing life.

The "pressure-relief" system of greasing is accomplished by running the motor at full speed, taking out the drain plug, and pumping in new grease until the new grease appears at the drain opening. In cases where overheating has been noted, it has been found that the plug was not left out long enough for the proper purging of any excess grease. Leaving the plug out for ½ hour while the motor is running is usually sufficient.

Another excellent system, also the pressure-relief type, is where the grease enters the back side of the housing, goes through the bearing, and comes out at the drain opening at the bottom of the closure.

One large concern has an interesting system. Its motors operate 7 days a week 24 hours a day. Hundreds of them cannot be seen from one end of the year to the other. These motors are cleaned and the bearing housings packed one third full of grease once a year, with no grease added during that period. This system has proved satisfactory for several years.

Bearing housings with considerable grease capacity and with small drain holes provided, do not lend themselves to this system of pressure-relief greasing. While insufficient pressure is exerted or developed to force grease through the small drain hole, enough pressure can be built up to cause a fluid-friction temperature rise within the grease pack itself.

Unless the excess grease can quickly purge itself through the shaft opening, a bearing failure might be encountered. The best way to service such a motor is to fill the housing manually about one third full at the start and add a small amount of grease every 6 months, never allowing the housing to become more than half full. It would not be amiss to attach a record tag to such motors with instructions on the quantity of grease to be added at each interval, the date of the application, and the initials of the man performing the work. The "guess" should be taken out of this problem as completely as possible.

There will almost always be a slight temperature rise noted immediately following the use of the pressure-relief system but it is of short duration and of no consequence. Higher-viscosity oils will sustain a temperature rise longer.

It is not necessary to grease motors operating under normal conditions oftener than once in 6 months.

Maintenance and Overhaul.

It is generally conceded that laboratory conditions do not prevail on plant operating floors or in repair departments and that conditions under which maintenance men must work are often very difficult. These conditions must be met with added care, not with added carelessness.

Careless handling of bearings during overhaul and poor maintenance practice can cause no end of trouble. The Annular Bearing Engineers' Committee has treated this subject fully in a recent publication.* It is with this in mind that every effort is being made to give it wide distribution.

The maintenance of antifriction bearings is not necessarily a difficult problem provided lubricants are selected with care and the men applying them perform their duties with skill and interest. It is the violation of ordinary good judgment which produces the most trouble. Difficulties need not reach epidemic proportions before the maintenance problem warrants increased attention. It is the ever-present monthly or yearly replacement record that will bear analysis. Some surveys show conditions that could be so much improved that one wonders what keeps the equipment in operating condition. Many plants, both large and small, could and should improve their methods, materials, and personnel for handling this important work. Bearing losses alone are not the only phase for consideration, since the effect of improper maintenance on other parts of the equipment must also be taken into account.

Overgreasing. The introduction of too much grease into an antifriction-bearing housing will always result in excessive temperatures which cause grease leakage which endangers electrical life of the motor, or causes bearing failures directly attributable to the elevated temperatures. Even though neither of the first two conditions results, the rate of oxidation is almost invariably increased which will necessarily decrease the life of the lubricant.

Undergreasing. Undergreasing consists of applying an insufficient quantity of lubricant or extending the period of relubrication over too long a time. In either case, this can only result in bearing failures.

Careless Handling of Bearings. Careless handling of bearings and introduction of foreign matter or dirt into them during overhaul impose a severe duty on the lubricant, cause undue wear and premature failure

Incorrect Assembly Methods. Incorrect assembly methods during overhaul result in damage to the bearings, cause rough or poor operation in the motor or generator and shorten bearing life. Men involved in this work should be asked to study the Annular Bearing Engineers' Committee instructions.

* "Anti-Friction Bearing Maintenance," Publication AFBMA-100, Anti-Friction Bearing Manufacturers Association, Inc.

Preventive Maintenance.

Cleaning and Flushing Materials. There are several high-grade cleaning and flushing materials under various trade names on the market today, prepared for this purpose. They are, in general, mineral spirits with sufficiently high flash points for safe use as a general-purpose solvent

Kerosene and commercial carbon tetrachloride are likely to be contaminated and may do more harm than good. If not thoroughly washed out, carbon tetrachloride can hydrolize to hydrochloric acid with consequent severe corrosion.

Cleaning Without Disassembly. Many motors are now designed with drain plugs at the bottom of the end caps which permit cleaning without disassembly. If equipment is found without drain openings at this point and it is possible to drill the end caps so they can be used effectively, it would be well to perform this work at an opportune time so that this cleaning process can be used.

Remove the pressure nipple, loosen the drain plug, and with the use of a small funnel pour in the solvent until it begins to bubble at the top. Run the motor for about 2 min. and drain. It might be necessary to loosen any hardened grease away from the drain hole to permit flow. Repeat this until the cleaning fluid comes out clear. Fill the housing with light oil, be sure to drain off the oil thoroughly, and apply grease.

The average bearing housing should be cleaned out at least every 3 years.

Disassembled Bearing Cleaning. Proper equipment should be provided for this work and should consist of at least two pails, each with a wide-mesh screen soldered about 3 or 4 inches from the bottom. The screen keeps the bearings from dropping to the bottom and sediment is not easily disturbed by cleaning-material turbulence. One pail will contain the solvent and the other, light oil.

For the most satisfactory results, a third pail should be added containing oil heated to about 180 F. This insures better protection against all contamination and moisture.

If the bearings are to be stored for a long time or held for spares it is best in addition to dip them in hot petrolatum or vaseline.

High-Temperature Operation. Development of greases for use in high temperatures over long periods has not been encouraging thus far. A high rate of oxidation develops after 180 to 190 F. is reached. It should be kept in mind that the rate of oxidation doubles for every 15 degrees F. temperature rise. This is definitely noticeable beginning at 140 F.

Advertising has been noted that some particular grease "will never deteriorate" with the inference that it will take about anything you want to give it, and we also have noted publicity that particular greases can be used up to 300 F., with no mention of time limits.

These statements are apt to mislead and confuse designing engineers planning new equipment, or operating engineers when planning new installations.

High melting points have nothing to do with oxidation values and are not difficult to obtain. The oil used in the manufacture of greases is highly important as to its oxidation and evaporation values.

All greases will eventually deteriorate even under normal conditions, and we have yet to see any grease that will stand 300 F., or even anywhere near that temperature for any appreciable length of time, regardless of speed. Without even putting grease into a bearing, anyone can place the lubricant in a heat-controlled oven in glass dishes with no catalyst present at 250 F. and observe the changes that take place. Weekly or even daily observations will show the increasing rate of destruction, the gradual change either to liquids or rubberlike substances. The rate of oxidation in this static test will be accelerated in actual operation.

In applying motors to equipment every effort should be made to keep the temperature variable as low as possible with the use of extended shafts, coupled drives, good ventilation, etc. As the heat variable rises, the oxidation-rate penalty increases.

If the correct grease consistency and housing design can be combined, the pressure-relief system can be used for maintenance regreasing, otherwise the housing should never be allowed to become more than half full. This can be accomplished by the initial application of a one-third filling, adding a small amount of grease every 3 months.

Housings should be cleaned out at least once a year.

Low-Temperature Operation. Several excellent greases have been developed for low-temperature operation in the lithium-soap-base class. These permit operation on even intermittent service at minus 60 to 70 F. It is claimed that some of these products will give satisfaction as low as minus 100 F.

In our northern plants using outdoor equipment or with motors installed in unheated buildings, it will be found that a good general-purpose ball-bearing grease with an oil viscosity of about 300 vis at 100 F., will be most satisfactory to minus 20 F.

Motors in Contaminated Atmospheres. If dust is the problem it will be very helpful to use a heavy unfilled sealing grease with a worked penetration of about 200 applied to the grease grooves so that only shaft clearance through the grease is permitted. This will bring the shaft clearance down from the usual 0.015 inch to about 0.003 inch.

If much product dust is actually getting into the bearing housing it might not be safe to use solvent-type flushing oils for cleaning without disassembly. The dust might not be of a character that would stay suspended in the solvent and rapid precipitation might cause the contamination to pile up in a solid mass at the bottom of the bearing or housing and fail to move out through the drain opening. It would just be left for the new grease to pick up in such large solid groups that an early bearing failure might result.

This problem should be studied on the job itself. It might be possible successfully to use an S.A.E. 20 oil so that the dust will float in it long enough to come out with the cleaning oil. This heavier oil will not produce the usual cleaning effect of solvents but it might be very useful and helpful.

The safest method, provided the housings are adapted to it, would be to use nothing but the pressure-relief system of greasing. Experience will have to determine how often it will be necessary to disassemble the unit for proper cleaning.

If destructive gases or vapors are present, such as sulphur dioxide or chlorine, it is best to try to use the pressure-relief system so that the housings are 85-90 per cent full all the time. Grease may have to be applied every month, or according to the percentage of atmospheric concentration or observed rate of destruction.

Bearing Types for Special Consideration.

Heavy loads are always involved with the use of roller and double-row ball bearings. In addition, particularly in roller bearings, there is considerable rubbing and sliding contact which requires positive and "wet" films working on all surfaces. It is therefore obvious that greases on the softer side should be used, providing films heavy enough to carry the loads but thin enough so that film shear at points of rubbing contact will not occur. The grease pack should not churn.

Figure 2 shows the SKF spherical roller bearing wherein it is essential that the inside ends of the rollers and the centerguide flange, as well as

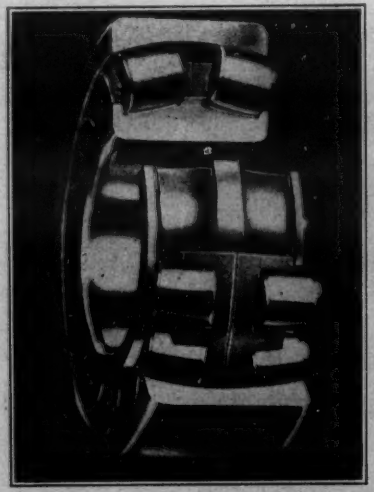


FIGURE 2.—SPHERICAL ROLLER BEARING.

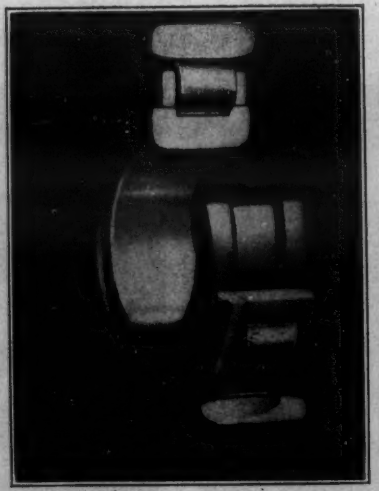


FIGURE 3.—CYLINDRICAL ROLLER BEARING.

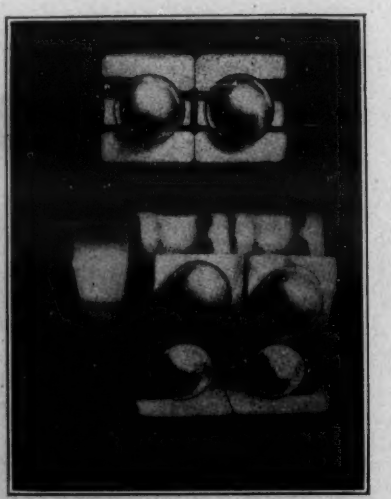


FIGURE 4.—DOUBLE-ANGULAR-CONTACT BALL-BEARING ASSEMBLY.

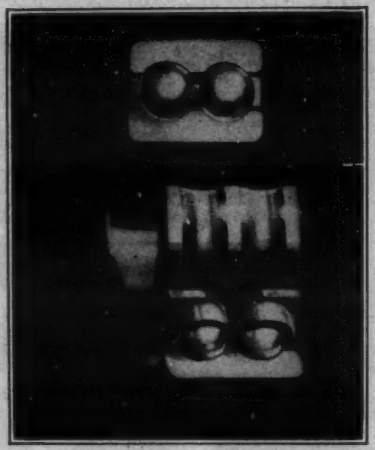


FIGURE 5.—DOUBLE-ROW DEEP-GROOVE BALL BEARING WITH CAST-BRONZE RETAINERS.

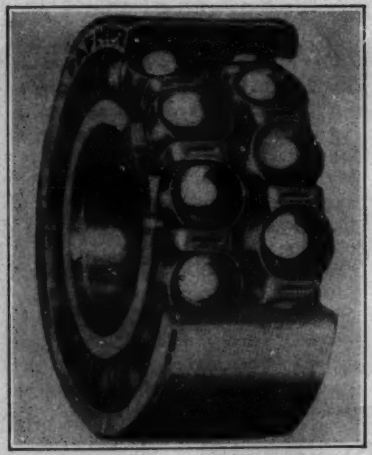


FIGURE 6.—DOUBLE-ROW DEEP-GROOVE BALL BEARING WITH PRESSED-STEEL RETAINERS.

the cage pockets, be well lubricated. While oil is the preferred lubricant, grease is being successfully used.

Grease for this type of bearing should have a worked consistency of 300 to 320. All rollers should be grease-smeared during assembly and both sides of the bearing packed with grease up to the bottom of the inner race. A small amount of grease can be added every 6 months. Quantitative greasing records should be kept, attached to the equipment or available from files at date of greasing for the man doing the work. Three years of operation should not see this bearing housing much more than one third full.

Every 3 years, or according to severity of operating conditions, the bearings should be removed, thoroughly cleaned, and the initial operation of applying lubricant repeated.

Figure 3 shows the cylindrical roller bearing where the two guide flanges as well as the roller pockets require lubrication. While the pressure-relief greasing system can be used with the proper housing design, it is best to proceed with the system outlined for spherical-bearing maintenance, using the same grease consistency, 300 to 320 worked penetration.

If drain plugs are installed at the bottom of the housing, these bearings can be cleaned, using the suggested method as outlined in this paper.

Figure 4 illustrates a double-angular-contact ball-bearing assembly, one of the three mounting setups that are used, back to back as shown, face to face or in tandem, according to the character of the load.

Many failures have occurred with double-row ball bearings because grease was applied to the outside row only, with the expectancy that sufficient grease would carry over to the inside row. Repeatedly, inspected failures have shown lack of lubrication on the inside row due to this method of greasing.

Grease for these bearings should be of at least 275-300 worked penetration; if it is much stiffer there will be considerable tendency for the lubricant to ride the retainers, offering too little lubricant support for the bearings.

If housings are built for low grease capacity and the drain plug is at the bottom, these bearings can be greased with the pressure-relief system, otherwise when mounting, both ball rows should be smeared with grease and both sides of the bearing filled one third full with a small amount of grease added every 6 months. Three years should not see these housings more than half full.

Proper cleaning of double-row ball bearings by the flushing method might be questionable; it would be better to remove the bearing housing. Thoroughly clean and repeat initial method of lubricant application every 3 years or according to severity of operating conditions.

Figures 5 and 6 illustrate the double-row deep-groove ball bearing, one with cast-bronze retainers, and the other with the pressed-steel type.

There are a great number of these bearings of the larger sizes in the field, the majority of them in housings with considerable grease capacity. A 275-300 worked-penetration grease should be used with an initial filling one third of housing capacity, packed on both sides. The balls should be grease-smeared during assembly. A small amount of grease can be added once in 6 months, never letting the housing become more than half full. Quantitative greasing records should be kept for such equipment.

For the smaller sizes, where grease capacity in the housing is small with drain plug at the bottom, satisfactory greasing is possible with the pressure-relief system.

Rust and Corrosion Problems. These problems have held the interest of ball-bearing and operating engineers for years, and they offer many

angles for study. Several basic reasons are offered for consideration:

1. Improper application of motor types.
2. Use of the wrong type or low-quality lubricant.
3. Improper cleaning of bearings.
4. Improper flushing of housings without disassembly.

Many surveys have shown that open-type ball-bearing motors have been used where splashproof or totally enclosed motors should have been installed. These have been found on roofs of buildings without proper housing protection, in food-processing plants where it is necessary to "hose down" every day, and in locations where heavy steam vapors exist. The operating locations cannot be changed but the application of motor types better suited to these conditions would offer considerable improvement.

The cheaper greases will almost invariably promote rust and corrosion owing to their original water content or rapidly developed organic acids.

A change from waterproof grease to water-soluble types would offer great improvement where rust or pitting is noted.

Waterproof greases isolate moisture so that it collects at the bottom of the bearing and housing or is interspersed in isolated globules through the grease. Every chance is thereby offered for clear water to reach the steel surface itself especially during periods of shutdown.

Soda or soda-lime greases absorb moisture-forming emulsions that provide protection against rust. If the absorption rate is severe, greasing periods might have to be shortened in order to correct any thinning of the lubricant. Here, again, the pressure-relief greasing system will be of great help and value, as any excessively softened grease can be eliminated through the drain opening. This is suggested procedure where moisture conditions are more or less constant. However, if water should get into a housing containing water-soluble greases under some temporary condition, all grease should be cleaned out and renewed.

Paper mills and textile plants offer excellent examples of production under conditions of high relative humidity. Surveys of conditions in these plants have shown, time and again, the telltale reddish-colored greases which were once amber-colored, the color change being caused by formations of iron oxide as a result of using waterproof grease.

Cleaning materials for disassembled bearings are sometimes left too long in buckets before renewal. Contaminated cleaning solvents or flushing oils are an invitation to rust.

In cleaning housings without disassembly, all traces of solvent should be removed with a light flushing oil. If left in the housings, many solvents can promote rust.

The use of compressed air is never recommended.

Vertical Motors. The same general practice outlined can be used for applying greases to the various designed housings, but in some cases it might be found best to use a grease with a worked consistency of 225 to 250. A change can be made from the general-purpose material if much leakage is noted, provided the reason is not improper greasing or overgreasing.

Slow-Speed Motors. It has been noted that slow speeds have a tendency to thin out a grease whereas high speeds stiffen it. If any thinning-out trouble is experienced, the housing, held to about a one-third filling, will help considerably. If the problem is particularly troublesome due to air emulsions, a 225 to 250 worked-penetration grease can be used effectively, never allowing the housing to become more than half full.

Aircraft Generators. Military aircraft during wartime conditions have placed a high premium on savings in weight in all accessory apparatus. With the advent of electrically operated and controlled aircraft equipment, motors and generators have been designed for high output with low weight. This in general means that the motors and generators must

operate at high speeds and high temperatures. The introduction of high altitude and its low temperatures, and also the widely separated geographical locations in which this equipment has to operate, have imposed extremely severe conditions under which antifriction bearings and their lubricants must operate. A plane does not always fly in one position, and this makes necessary versatility in mounting the apparatus.

Much research must be performed in order to find the answers to all the problems existing even though lubrication research has done much to improve the operation of aircraft equipment. Progress made in the aircraft field will necessarily be reflected in the industrial applications of the future. Lubrication development for conditions as outlined is a "must" project of the present and future.

Grease-Quality Control. Pertinent to this entire problem is the necessity of getting uniform material from the lubricant manufacturer month after month. Experimental kettles might give us an interesting product but the production kettles many times spoil the picture. Regular production batches have been known to vary considerably. Therefore, while a lubricant manufacturer might produce an interesting grease, it becomes his problem to prove that he can manufacture the same superior material over a long steady period. It should and it does require a long time to build real confidence around a ball-bearing lubricant.

Handling Equipment and Storage. Management has the responsibility of providing the best possible handling equipment for all lubricants. Nothing will encourage the greasing and oiling personnel more. No matter what the initial cost, results will prove the investment well worth while. Good example starts at the top.

Provision must be made for proper storage facilities which make it possible to keep all lubricants clean. A room should be specially provided for this purpose, planned so that containers in use are easily accessible. It should be possible to clean thoroughly and easily at any time.

If lubricants of necessity are further distributed through the plant, storage facilities, such as steel cabinets, should be provided.

Personnel. Men chosen for the work of applying lubricants should be hand-picked. Some men are naturally clean and neat as well as methodical. If this type can be found it would be well to place them in this group. Sloppy men do sloppy work and the care of expensive equipment should not be placed in their hands.

They should have mechanical training. During the work of applying lubricants a good mechanic will often detect mechanical trouble before it becomes serious. Trouble caught in its early stages might cost a few dollars for adjustment but if not noticed, would eventually cost a considerable sum.

Too many instances are noted in which this type of work is handed to the men with the lowest hourly rate. Better judgment than this should be practiced.

Conclusion.

This is a general over-all picture of the many angles involved in the problem of lubrication of ball-bearing motors. Any phase warrants considerably more treatment than is offered in this paper. It is hoped that a great deal more material on this subject will be forthcoming.

Ball and roller bearings are today considered to be a critical part of the nation's war-effort stock pile. Improper maintenance that results in bearing-replacement withdrawals from this stock pile definitely calls for study and great interest on the part of everyone concerned.

The cost of the replacement bearing itself is only a small part of the cost in lost production and the consumption of valuable man-hours needed to correct the results of improper maintenance which in most cases, could be avoided.

Close study of shop repair orders and the equipment coming in for overhaul would undoubtedly reveal much laxity in the application of proper practices in lubrication-preventive maintenance.

Since we are all interested in extending the life of antifriction bearings to the maximum years of service, we must work together to this end; the bearing manufacturers, the equipment builders, the lubricant producers, and the eventual users.

ACKNOWLEDGEMENT.

The author wishes to acknowledge the co-operation and counsel of many research and production engineers of electric-motor manufacturers, as well as the engineers and metallurgists of ball-bearing manufacturers, over a period of several years. It has all been greatly appreciated. Figures 2 to 5, inclusive, are published through the courtesy of SKF Industries, Inc., and Figure 6 is published through the courtesy of Fafnir Bearings, Inc.

SHOP NOTES ON WELDING STAINLESS STEEL.

This article by Vincent J. Shanahan is reprinted from the August 1945 issue of the "Welding Journal". It is a practical discussion of the factors with which shop personnel who weld stainless steel should be familiar.

Maintenance, repair and fabricating shops will utilize increasing amounts of stainless steel in the future. Such shops do not always possess precision welding equipment as Unionmelt machines and elaborate spot-welding jigs or other devices so frequently referred to in the literature on stainless steel. What shop foremen and welders can use is a practical summary data for stainless fabrication.

What type of rod should be used? What size? How much will we need? Will the work distort? What does heat do to stainless steels? These are the questions discussed in the five sections of this article.

1. Comparison of welding carbon steel and stainless steel.
2. Effect of heat on stainless steel.
3. Selection of the type of electrodes.
4. Selection of size and quantity of electrodes.
5. Operating instructions.

I. COMPARISON OF WELDING CARBON STEEL AND STAINLESS STEEL.

Many welders who have successfully fabricated straight carbon steel are reluctant to weld stainless steel and have mistaken ideas of the difficulty of working stainless. First, it should be made clear that the 18-8 stainless type can be very readily welded by the metallic arc and the acetylene processes.

To obtain clean, ductile, uniform welds, we should know the general classification of stainless in comparison with carbon steels.

If carbon steel is represented by the number 100 for each of the following classifications, we have these approximate values for 18-8 stainless:

Specific electrical resistance	640
Melting point	95
Specific heat resistance	110
Thermal conductivity	33
Coefficient of expansion	145

Briefly, we see that 18-8 has six times the resistance to electrical current and melts at lower temperature, conducts heat only one-third as fast and

expands half again as much. This indicates that allowances must be made for expansion and contraction, and that holding fixtures would be desirable in many cases, and that chill bars and similar devices should be carefully considered in setting up the job.

The fact that the electrical resistance of 18-8 stainless- is about six times that of mild steel means that lower currents can be used. The tables in "Operating Instructions" will indicate these voltages. Straight chrome grades of stainless expand about 10 per cent less than carbon steel and hence welded structures can be designed in almost the same manner as mild steel. The low thermal conductivity (approximately one-half mild steel) may cause slight warpage in gages and suitable jigs are advisable. The electrical resistance of straight chromium grades is from three to six times that of mild steel with only a slightly lower melting point. The straight chromium stainless steels may be divided into two groups; those hardenable by heat treatment, and those non-hardenable. The latter group (Types 442, 443 and 446) contain 18 per cent and over of chromium and heat tends to cause embrittlement, which cannot be entirely relieved by subsequent heat treatment. This grade is not generally used in applications subject to vibration, shocks or deformations.

The straight chromium stainless steels from 10-18 per cent (Types 410, 420, 430 and 440) are hardenable and air-hardened in cooling from the welding temperatures. Heat treating and annealing from 1350 to 1400 degrees F. will increase toughness and ductility of the weld area.

II. Effects of Heat on Stainless Steels.

The fundamental principle of all welding, whether by electric arc or by flame, is to apply sufficient controlled heat at a given point to melt and fuse the section to be welded. The above principle holds true, whether a filler rod is used or not.

Heat, then, is the most important factor, not only because of the distortion of stainless under temperature, but also because high temperatures definitely affect the ultimate corrosion resistance or physical properties of the stainless. Certain techniques and alloys have been developed to counteract the heat of welding.

Let us consider the chromium-nickel stainless steels first.

In welding 18-8 steels, the area on both sides of the weld is subject to varying degrees of temperature, ranging from approximately 2900 degrees F. at the weld to room temperature at the extremities of the sheet or plate on a large piece. In the heated area there is a band at which temperatures between 900 and 1600 degrees F. exist for a period of time.

Within these temperatures the carbon in the steel combines with the chromium to form chromium carbides along the grain boundaries. This process is called "carbide precipitation." Such an area is more subject to corrosion than the base metal, because the chromium content of steel at that point is lowered by the precipitation.

Several things can affect the degree of formation of the carbides. If a low-carbon stainless (Type 304) is used, the precipitation is less. A low-carbon (Type 308) electrode is also of great assistance in controlling the carbides. The elements columbium and titanium (Types 347 and 321) may be added to stabilize the stainless and to prevent the formation of chromium carbide. The length of time the steel is held between the critical range of 900 to 1600 degrees F. has a direct and harmful bearing on the amount of carbides formed. The ratio of the alloy content to the carbon content affects this precipitation.

Heat treatment of the entire welded article is the best way of eliminating the carbides, assuming that a stabilized stainless is not being used. After the carbides have formed at some point over 900 degrees they cannot be reabsorbed in the solid austenitic condition of the steel

IRON AND STEEL INSTITUTE TYPE NUMBERS FOR STAINLESS STEELS.

Steel Type No.	*Carbon*	*Chrome*	*Nickel*	*Other Elements*	*Foe Welding Use Electrode Type Numbers*
302	0.08–0.20	18.00–20.00	8.00–10.00	Mn 1.25	308
303	0.20 max.	18.00–20.00	8.00–10.00	S or Se 0.07 min., or Mo 0.80 max.	308
304	0.08 max.	18.00–20.00	8.00–10.00	Mn 2.00 max.	308
309	0.20 max.	22.00–26.00	12.00–14.00	.....	309
310	0.25 max.	25.00–26.00	19.00–21.00	.....	310
316	0.10 max.	16.00–18.00	14.00 max.	Mo 2.00–3.00	316
321	0.10 max.	17.00–20.00	7.00–10.00	Ti (min) 4 × carbon	347
330	0.25 max.	14.00–16.00	33.00–36.00	.....	330
347	0.10 max.	17.00–20.00	8.00–12.00	Cb (min) 10 × carbon	347
410	0.15 max.	10.00–14.00	.....	.....	410
416	0.15 max.	12.00–14.00	.....	S or Se 0.07 min., or Mo 0.60 max.	410
420	Over 15	12.00–14.00	.....	.....	430
430	0.12 max.	14.00–18.00	.....	S or Se .07 min., Mo. 60 max.	430
440	Over 0.12	14.00–18.00	.....	.....	446
502	0.10 max.	4.00– 6.00	.....	Mo. 50 max., Cb. 50 max.	501

until an annealing temperature of 1800-1900 degrees F. has been reached. On cooling from these higher temperatures either in air or water carbides are again dissolved into "solution" in the metal. There are, to be sure, many mildly corrosive or sanitary applications where the presence of carbides is not harmful and no effort need be made to control them.

Carbide precipitation can be lessened by the shooting of a stream of cool water over the weld after laying each rod of weld. If the plate or sheet is thick enough to require an inside and outside weld, the last weld to be made can be cooled by spraying water on the opposite side of the seam. However, more desired results can be attained by spraying water on each side of the seam as it is being welded.

Caution must be taken not to get water into the weld crater at the time of welding. This is a very practical application on units that are too large for heat-treating ovens.

Since many fabricated sections cannot be readily heat treated, metallurgical research developed the use of two other alloying metals, titanium and columbium, which prevent the formation of carbides when added to the chromium nickel steel in fixed proportions to the carbon content. The most common of these stabilized steels are Type 321 (titanium) and Type 347 (columbium). Columbium may also be added to other chromium steels (Type 309 Cb and Type 310 Cb) in order to prevent carbides.

Titanium is never used in stabilizing electrodes because the element oxidizes almost completely at welding temperatures. Columbium is always the stabilizing element in 18-8 stainless electrodes.

The straight chromium grades do not lose corrosion resistance as a result of heat of welding. They are stable at all temperatures. The physical properties of these steels are decidedly affected by heat, and they are as a group more difficult to weld. Heat enlarges the grain size of the chromium steels.

In general, those under 18 per cent chromium (Types 410, 420, 430 and 440) air harden after welding, lose ductility and thus become brittle. The strains which are created by the heat of welding can be so great as to cause the metal to crack, usually in the area adjacent to the weld. If the metal section can be preheated up to a range of 350 to 400 degrees F. before welding, the temperature changes are not so abrupt, but are spread over a larger area stresses a better opportunity to relieve or spread themselves. Postheating up to temperatures of 1200 to 1400 degrees F. may be useful in some cases.

III. Selection of the Type of Electrodes.

In general the type of electrodes used should be the same as the type of metal to be welded. The analysis of the rod is usually of higher alloy composition in order to insure a weld equal in analysis to the parent

Analysis of Electrodes.

Type	*Carbon*	*Chromium*	*Nickel*	*Molybdenum*
308	0.07% max.	19% min.	9% min.	Mn 2.00
309	0.25% max.	25% min.	12% min.	
310	0.25% max.	25% min.	20% min.	
316	0.10% max.	18% min.	8% min.	2–4% Molybdenum
330	0.25% max.	15% min.	35% min.	
347	0.07% max.	18% min.	8% min.	with Columbium Cb 10 × Carbon
410	0.12% max.	12% min.	.....	
430	0.12% max.	16% min.	.....	
446	0.35% max.	23% min.	.....	
501	0.10% max.	.....	.....	2% with Molybdenum

metal. The coating of electrodes provides a flux which shields the molten steel from the arc, prevents oxidation of the chromium, helps to stabilize the arc and gives a more corrosion-resistant weld. In general a higher alloy content is used in the electrodes rather than in the material being welded. For example 18-8 electrodes are frequently being used to weld straight chromium steels. Type 302 can be quite successfully welded with Type 308 electrodes although Type 347 would be even better for the more severe corrosive applications. The use of electrodes with a different alloy content than the material to be welded should be carefully checked to determine if the corrosive conditions to be encountered would permit such a change.

IV. Selection of Sizes and Quantity of Electrodes.

To assist the welder to determine the size and quantity of electrodes needed for a given job, the following tables can be used as a general guide. Naturally special conditions may alter these figures somewhat.

Butt Welds.

Plate Size	*Rod Size, In.*	*Lb. Rods per Ft. Weld*	*Current, Amp.*	*Arc Volts*
30–22 gage	1/32	0.04	8– 15	18
18–20 gage	3/64	0.06	15– 25	20
14–16 gage	1/16	0.10	40– 50	22
3/8 In.	3/32	0.18	80–100	24
5/16 In.	1/8	0.30	110–130	26
1/4 In.	5/32	0.50	150–170	27
3/8 In.	3/16	0.75	180–200	28
1/2 In.	1/4	1.20	225–250	30
3/4 In.	5/32–1/4	2.75	225–250	30
1 In.	3/16–1/4	4.50	225–250	30

Fillet Welds

Plate Size	*Rod Size, In.*	*Lb. Rods per Ft. Weld*	*Current, Amp.*	*Arc, Volts*
30–22 gage	1/32	0.03	7– 12	17
18–20 gage	3/64	0.05	12– 20	19
14–16 gage	3/64	0.08	15– 25	20
3/8 In.	1/16	0.16	35– 45	21
5/16 In.	5/32	0.27	70– 90	23
1/4 In.	1/8	0.45	100–120	25
3/8 In.	5/32	0.65	140–160	26
1/2 In.	5/16	1.10	170–190	27
3/4 In.	1/4	2.00	220–240	29
1 In.	3/16–1/4	4.10	220–240	29

Coated electrodes must be kept dry. This can be attained by heating rod storage room or cabinet to about 80 or 90 degrees F. Poor welds result in a damp coating. Damp coating will crack and drop off while welding and the bare electrode must then be discarded. Standard lengths of rod are 14 to 15 inches for 1/8 to 1/4 inch diameter and 9 inch for 1/32 to 3/32 inch. Center grip electrodes are 18 inches long.

V. Operating Instructions.

The following practical rules for stainless welding conscientiously followed, should improve both quality and production.

1. First clean the work to be welded. Use stainless wire brushes and stainless steel wool. However, under wartime restriction this can be substituted for clean wire brushes. Keep the welding benches and fixtures free from all dirt of any type.

2. Align all edges very exactly. Wherever the work will permit, holding fixtures should be used. If jigs are not feasible or economical, tack welds are advised. The tacking should be on the opposite side of the weld if possible. The size and number of tacks should vary with the type of work and weight of material, bearing in mind that the tack should fuse in with final weld or the tack should be ground or chipped out depending on both the quality of tack and quality of work desired. Chill bars, placed on the weld to convey heat away quickly, will help to minimize the carbide formation. Copper chills are preferred.

3. Reverse polarity is generally used to weld stainless, the electrode being positive and the work negative.

4. A short arc produces the cleanest weld with the least splatter and best penetration. Do not let the arc become so short that the coating touches the metal.

5. To start, strike the arc just ahead of the crater, then move back slowly to remelt the crater.

6. Advance at a rate which will completely fill all undercut, and at a rate which will not allow any slag inclusion.

7. Clean off each weld bead thoroughly before a new one is laid. Any oxides or residues of flux on the steel will be focal points of weakness, both physically and chemically. Electrolysis may start within such pockets of impurities.

8. Avoid weaving the weld unless necessary on very wide beads. The motion tends to make slag inclusions. However, weaving can be accomplished after a little experience but caution is very necessary.

9. One pass can be used to weld thickness up to 3/16 inch. Over that multiple passes are superior.

10. Renewing the electrode permits the metal to cool and solidify. It is very necessary that the old crater should be clean before striking the new arc.

11. Clean the welded surface to resist corrosion, as well as improve appearance. The heat discoloration can be removed by pickling or grinding. Weld spatters must be chipped off with a chisel or sharp tool. They are the starting point of corrosion and destroy appearance.

12. To shear or punch stainless steel requires about one and one-half or two times the capacity of mild steel. To prepare welding edges, planing is recommended, however, if this type of preparation is not attainable, grinding may be used. A caution must be exercised in grinding to keep the edges straight or parallel with each other, where neither type of preparation available or practical because of conditions. One hundred per cent penetration can be obtained by welding in a slight vertical weld uphand on metal up to eight gages. On gages lighter than 14 gage or 16 gage no edge preparation is necessary, but the larger rods, more heat and speed are obtained by welding on a slight vertical position down hand.

Oxyacetylene Welding.

Oxyacetylene welding will be referred to only briefly, since its greatest range of usefulness is in welding 20-gage and lighter stainless steel. The corrosion resistance, strength and appearance of the weld are best when a slightly reducing flame is used. A reducing flame (slightly

excess of acetylene) is characterized by a bright blue cone inside the flame with a slight feathering beyond the cone. An oxidizing flame should and can be avoided; being easily recognized as the smallest flame in over-all proportions. Uncoated or bare rods are commonly employed. A thin paste of flux may be brushed on the rod as the operator desires.

The objective of this article has been primarily to assemble some practical information for the shop welder on the fabrication of stainless steel. But this article is confined largely to "cause" and "effect." There is much more that could be added, most of it is highly technical in nature as in the case of Unionmelt, atomic hydrogen, Heli-arc and resistance welding.

Atomic hydrogen welding has come into the field of stainless steel welding which will be developed for smaller shop use. Atomic hydrogen welding can attain such a high degree of proficient fusion that it defies any physical or chemical analysis to detect the weld area. This type of welding can be used either with or without the filler rod.

THE PRINCIPLES OF RADAR.

This article by J. M. A. Lenihan, reprinted from the November 1945 issue of the "Shipbuilder and Marine Engine-Builder", gives a simple explanation of the principles of radar for the layman.

Information which has become available since the termination of the war indicates how great was the part played by radar in defending these islands during the early years of the European conflict and in carrying our offensive power into the enemy's territory during its later stages. Consideration of the peace-time uses of this invention is now being undertaken in many quarters and it may be no more than a year or two before radar apparatus becomes a standard item of equipment in every large ship and aircraft. The purpose of the present article is to explain the physical principles on which radar is based.

Radar is essentially a simple process. It involves the measurement of the (very short) time interval which elapses between the despatch of a radio signal and the receipt of its echo from a reflecting object. The velocity at which the signal is transmitted being known, it is possible to estimate the distance of the object, while determination of its angular position, in elevation and azimuth, may be made by the means of suitable devices at the transmitting and receiving stations. To appreciate the technique employed, it is necessary to refer to certain fundamental physical processes. These are discussed in the following paragraphs.

Electromagnetic Radiation.

A current flowing in a conductor generates a magnetic field the polarity corresponding to the direction of flow; if an alternating current is employed, the magnetic field must collapse after each half-cycle of the alternation before being re-established with reversed polarity.

This process results in a loss of energy, which occurs in the following way. As the current increases during the first quarter-cycle of the alternation, the magnetic field also increases, and lines of force spread outwards from the conductor. In the next quarter-cycle, during which the current falls to zero, the magnetic field begins to collapse. At this stage, an effect corresponding to inertia becomes apparent; for the field does not collapse completely before the current again begins to increase in the reverse direction, thereby putting out more lines. The result is that part of the energy represented by the magnetic field is not returned to the circuit which generates it, but remains permanently outside the wire and is,

indeed, pushed further out by the expansion of the field in each succeeding half-cycle of the alternation. Energy is thus released in the form of a magnetic field which is propagated outwards and which alternates with the same frequency as the current which produced it. An alternation of electric-field strength is necessarily associated with this alternating magnetic field and the two together constitute an ***electromagnetic wave*** which, in free space, moves at a speed of 186,000 miles per second, the speed in air differing from this value by only a negligible amount. Radio waves generated in this way are of the same fundamental nature as heat, light, X-rays and other forms of electromagnetic radiation from which they differ only in wave length and frequency. These two properties of a wave are related to the velocity of propagation by the relation:—

$$c = n\lambda,$$

where c = velocity, n = frequency and λ = wave length.

The wave lengths employed in those applications of radar which it is permissible to discuss are mainly in the range between 10 and 1000 cm., corresponding to frequencies between 3000 and 30 megacycles per second.

Propagation of Radio Waves.

Electromagnetic waves travel, in general, in straight lines, and may be totally or partially reflected at the boundary between two media. Radio waves are readily reflected by metals, and to a less extent by other solid and liquid materials. To transmit radio signals, it is necessary to have a system of valve circuits for generating an alternating current, and an aerial conductor from which radiation may occur. A single conductor behaves rather in the manner of a point source, radiating uniformly in all directions; but, with several conductors in an array, directional properties may be produced, resulting in the emission of a narrow beam of radiation.

Reception of the radio signals is effected by a conductor placed in their path. The alternating magnetic field produced in the space surrounding the receiving aerial induces in it an electromotive force, which alternates in the same way as the current in the transmitting aerial. Since the signal suffers attentuation during its journey (in accordance with the inverse-square law) and various other losses in addition, it is usually of very small amplitude on arrival at the receiver, and must therefore be amplified by further valve circuits before it can be recorded by any of the customary devices.

In radar, a powerful signal, equivalent in some cases to hundreds of kilowatts, is radiated. It is convenient to assume in the first instance that the transmitting aerial is without directional properties, so that radiation travels outwards in all directions. An object (an aircraft, for example) in the vicinity will reflect some of the radiation falling upon it; but, as the reflected signal also spreads in all directions, the echo picked up by a receiver on the ground is very slight indeed. By way of example, consider a transmitter radiating at a power of 1000 Kw. Calculation, based on the inverse-square law, shows that the amount of energy falling on a target presenting an area of 1000 square feet at a distance of 100 miles is about 0.6 joules per second; while the power conveyed by the echo to the receiver, with an effective area of, say, 10 square feet is 4×10^{-12} Watts. This figure is near the limit below which a receiver will not operate satisfactorily, because, for example, voltages comparable with that which a signal of this intensity can induce in a receiving aerial are generated spontaneously by thermal agitation of the molecules of the aerial itself, and by other effects of a like nature. Improvement may be obtained, with a corresponding increase in maximum range, by the use of aerials having directional characteristics at the transmitter and receiver; and, with such aerials, it is necessary to "scan" the region in which a target may appear by continuous rotation of the aerials.

Modulation.

Certain difficulties are encountered in the endeavor to give practical expression to the ideas outlined in the foregoing paragraphs. The echo having been produced, a reasonably accurate method of measuring the time interval elapsing between the despatch of the original signal from the transmitter and its arrival at the receiver must first be found.

It has been explained that, before energy can be radiated from a conductor, an alternating current must flow in it. Now, one essential feature of an alternating current is that each cycle is identical with those preceding and following it. If, therefore, a continuous wave be transmitted, the echo has no characteristics whereby it may be distinguished from the transmitted signal. The problem is analogous to that encountered in the attempt to find the distance of a cliff by timing the echo of a whistle. If the whistle be sounded continuously, the echo becomes confused with the original sound, so that it is not possible to separate them nor to make an estimate of the time taken. In these circumstances, it is necessary to emit a sound of such short duration that a reasonable time elapses between the end of the signal and the receipt of the first part of the echo. In other words, the transmitted signal is discontinuous and each echo received may be associated with a particular part of it.

A solution along these lines has been successfully adopted in radar, though, naturally, certain modifications in detail are required. It is not practicable to radiate short bursts or pulses of energy by supplying corresponding pulses to the transmitting aerial, for the following reason. The time taken for a signal to travel to a "target" at a distance of 100 miles—a reasonable figure for many types of radar equipment—is about

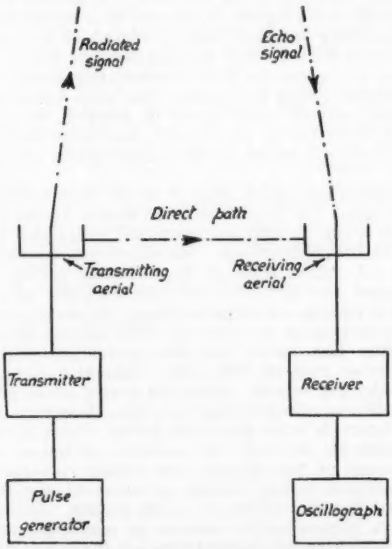


Fig. 1.—Schematic Arrangement of Transmitting and Receiving Equipment.

1/2,000 second and, allowing for the return journey, an interval of at least 1/1,000 second must elapse between successive pulses to avoid confusion. Now, radiation takes place most effectively from a conductor whose length is of the same order as the wave length to be transmitted; and, since the wave length corresponding to a frequency of 1000 pulses per second is 186 miles, it is not practicable to radiate any appreciable amount of energy directly in this form. The procedure adopted is to radiate at a high frequency, corresponding to a wave length of a few meters or less, so that aerial equipment of reasonable size may be used, the amplitude of the signal being varied or *modulated* by a momentary increase, applied, perhaps, 1000 times per second. In this way, the transmitted wave receives an easily recognizable imprint every thousandth of a second, so that the pulses may be identified in the received signal.

For greater ranges, the separation between successive pulses must be increased, since, to avoid ambiguity, the echo from one pulse must be received before the next pulse has left the transmitter. Details of the methods employed to generate the pulses or the high-frequency carrier-wave which supports them would be out of place here; and, in regard to the receiver, it is only necessary to say that the carrier wave disappears during the amplification and other processes undergone by the signal, so that the final output from the receiver is a series of pulses corresponding to those emitted by the transmitter, together with any which may have been reflected from objects in the vicinity. The transmitter and receiver are usually only a few yards apart, the general scheme being represented in Figure 1.

Timing Devices.

The methods involved in the production of the signal and the reception of its echo having been considered, it is necessary to examine the problem of measuring the minute time interval between the two pulses. This interval is never more than a few milliseconds, and may be very much smaller. The pulses may last for only a microsecond, and mechanical indicators are therefore ruled out, since it is not possible to construct a mechanical system with so little inertia as to permit it to respond to a phenomenon whose duration is as short as a millionth of a second. The difficulty is resolved by means of the cathode-ray oscillograph, whose function is essentially similar to that of the Duddell oscillograph familiar in electrical practice. The moving element consists not of a coil but of a beam of electrons, the movements of which are recorded in the form of a graph of the alternating quantity under examination—in this case the output from the radar receiver. The beam itself is, of course, invisible; but if it is allowed to fall on a screen coated with fluorescent material, the luminous spot produced indicates the displacement impressed on the electron "pointer."

The elements of a cathode-ray oscillograph are indicated in Figure 2. Electrons obtained from a heated metallic filament are concentrated into a narrow beam by the repulsion of a negatively-charged shield placed around the source. They are then accelerated by passage through two or three anodes, usually in the form of discs or cylinders, each with a small central aperture. The anodes are maintained at a positive potential relative to the filament and, when suitably designed, concentrate the electron stream further, so that a small bright spot is obtained on the screen. The beam next passes between two pairs of deflecting plates, XX and YY, arranged vertically and horizontally. A steady potential difference between the X plates deflects the beam horizontally in the direction of the positive plate; if the Y plates are used, a vertical displacement is similarly obtained.

The application of an alternating potential difference to either pair of

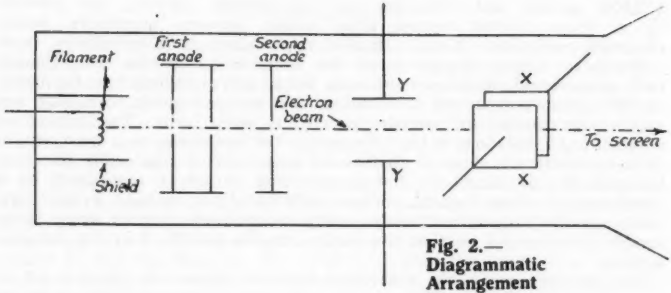


Fig. 2.—Diagrammatic Arrangement of Cathode-ray Oscillograph.

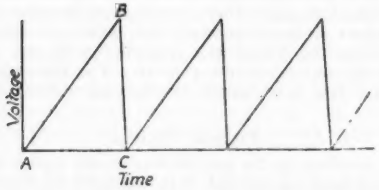


Fig. 3.—Wave Form of Voltage, to Time-base, as applied to the X-plates of the Oscillograph.

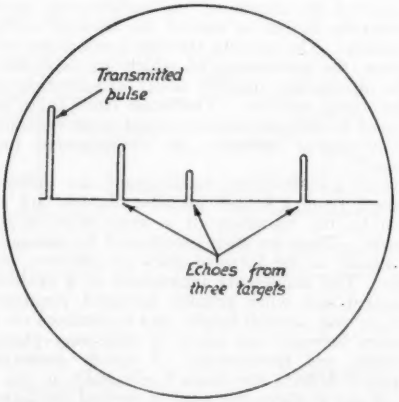


Fig. 4.—Sketch illustrating Radar Display on Oscillograph Screen.

plates causes the beam to oscillate to and fro, with the result that the luminous spot on the screen describes a line; and at frequencies greater than a few cycles per second the motion of the spot cannot be resolved by the eye, a continuous trace being observed. In the application with which these notes are concerned, a potential difference of the form shown in Figure 3 is applied across the *X* plates. The spot, under this influence, moves across the screen at a uniform rate and then returns rapidly to the starting point. This time-base, as it is called, is synchronised with the transmitter in such a way that the electron beam executes one journey across the screen and back in the interval between the transmission of two pulses. Suppose that, while the time-base is in operation, the output voltage of the receiver is applied to the *Y* plates of the oscillograph. At the beginning of the sweep, the transmitted pulse is picked up by the receiver and appears as a vertical line on the screen of the tube. Any reflected pulses which may reach the receiver appear on the screen at a later stage in the sweep, giving the appearance shown in Figure 4. After a thousandth of a second or so, the spot returns to its original position another pulse is emitted, and the processes just described are repeated.

The electron beam thus paints a rapid succession of pictures on the fluorescent screen; and, on account of the persistence of vision, a single trace is observed. If the reflecting object is in motion, the pulse returned by it moves across the screen in the appropriate direction; the distance on the screen between the transmitted pulse and its echo is proportional to the distance between the transmitter and the target.

Position-Finding.

To fix the position of an object, it is necessary to know its elevation and azimuth (or bearing), as well as its distance from the transmitter. The necessary angles are found by means of directional aerials, in which a number of conductors, of suitable lengths and relative positions, are assembled in an array so as to receive—or transmit—signals within only a narrow angle. In early radar equipments, the transmitting aerial was designed to "floodlight" the largest possible area of sky, while directional aerials were employed in the receiver. An aerial of this kind picks up signals with maximum efficiency when it is pointed directly at the source (or, in some designs, along a line perpendicular to this direction), and the receiver thus requires two sets of aerials—one for elevation and another for azimuth, each fitted with a scale so that the appropriate angles may be read. The process is, in detail, rather more complicated than this, but enough has been stated to indicate the principle involved.

In later radar equipments, the aerials of both transmitter and receiver have directional properties, so that a small portion of the sky is intensely "illuminated" with radio waves at any instant. The two sets of aerials are kept in step throughout their travel, and the arrangement is closely analogous to that of a searchlight used for visual illumination. To produce very narrow beams of radiation, short-wave lengths are needed, and this consideration has been largely responsible for the immense development of centimeter-wave technique which has been undertaken during the war. The technical efficiency attained in many applications of radar has been remarkable. Enemy aircraft have been detected and plotted while still hundreds of miles from our shores, bombers have been able to "pin-point" their positions with an error of less than fifty yards, while submarines have been detected at ranges of several miles by the reflections from their periscopes.

Conclusion.

The use of radar in merchant ships presents a number of problems It is, for example, extravagant in normal times to install equipment for the detection of aircraft; and though this was the original purpose of

radar, it will, presumably, find no place in the marine installations now being designed. It is also important to remember that, during the war, radar equipment has been produced in great haste, so that the methods and designs are not necessarily those which will be found suitable in the future. Moreover, many types of equipment possessing great range and sensitivity require expert maintenance which it may not always be practicable to provide on shipboard. It therefore seems likely that a specification emphasizing reliability of operation and ease of maintenance, with perhaps some sacrifice of range and sensitivity, will be appropriate for marine use. Until these and kindred matters have been decided, it is hardly possible to discuss in detail the construction or performance of radar equipment which may ultimately be adopted for merchant ships.

The value of radar as an aid to marine navigation is well illustrated by the following incident.

In March, 1944, a Japanese submarine was detected while maneuvering into position to attack the new Prince liner *Chinese Prince*. The captain turned his ship towards the enemy's position with his vessel at full speed. The enemy submarine immediately made a "crash" dive to a depth far below that at which she could launch an attack, while the *Chinese Prince* endeavored to avoid further action. However, the Asdic soon detected the enemy in pursuit, and Captain Thornton then headed his ship towards the shelter of a cluster of atolls through which, in the rapidly falling darkness, no submarine would dare to venture submerged, for the channel was intricate. The atolls were no more than a few feet above the sea surface, but they were covered with cocoanut palms, and the radio officer contrived to get a radar reaction from the palm trees perfectly clearly. In the darkness, the ship raced through a channel that would perhaps have deterred many navigators even in daylight. On reaching open water again, the *Chinese Prince* had shaken off her pursuer, and she proceeded on her voyage without further incident.

A MARINE GAS TURBINE PLANT.

This paper was prepared by C. Richard Soderberg, Ronald B. Smith, and Lt. Comdr. A. T. Scott, U.S.N.R., and was presented at the annual meeting of the Society of Naval Architects and Marine Engineers in November, 1945. Because of the significance of its contents to naval engineers, it is reproduced below in its entirety, except for the Appendix, for the benefit of those who do not have access to a copy of the "Proceedings of the American Society of Naval Architects and Marine Engineers".

In 1943 there was presented before this Society a theoretical discussion of the potentialities of the gas turbine with particular regard to its application for marine propulsion (1). Under the sponsorship of the Bureau of Ships a 2500-horsepower gas turbine, embodying these concepts, has been constructed by the Elliott Company, and the unit has been under test since the latter part of 1944. The object of the present paper is to give a progress report on the operation of this machine, together with the salient features of its design.

It was postulated at its inception that the Elliott-Lysholm gas turbine should be designed to deliver power at an efficiency economically comparable to that attained by steam turbines and Diesel engines. The development is significant because practically all other known attempts have been postulated on the simpler premise of low efficiency. Application to naval propulsion service imposed additional considerations with regard to compactness, simplicity, light weight, reliability and, particularly, high partial load efficiency and satisfactory maneuvering character-

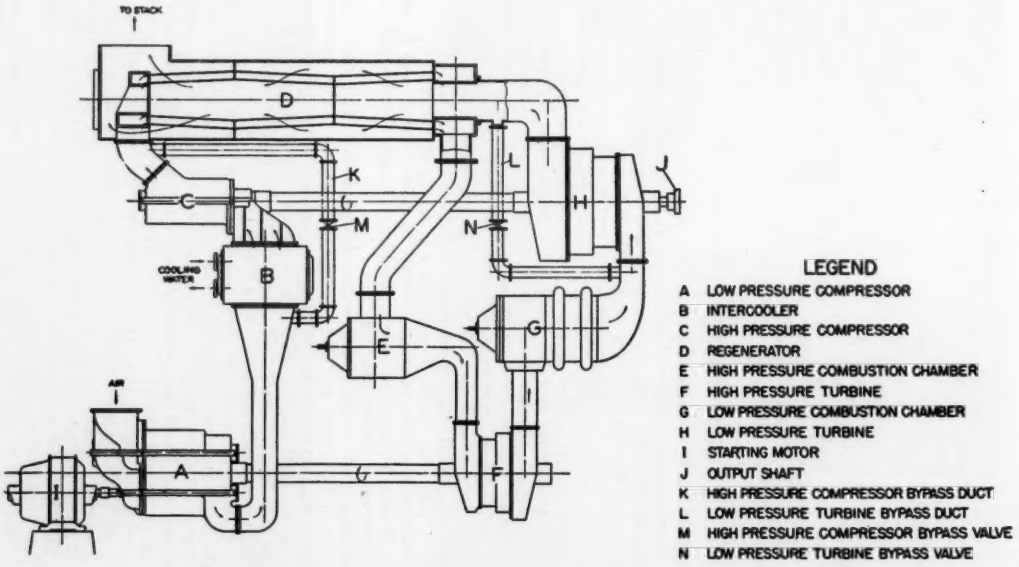


FIG. 2.—SCHEMATIC ARRANGEMENT OF THE ELLIOTT-LYSHOLM GAS TURBINE PLANT

istics. With regard to the last two requirements, the simpler gas turbine plants, upon which widespread attention has been focused in recent years, are notably deficient.

With the materials available in 1942 it was evident that a plant designed to operate at approximately 1200 degrees F. could have an expected life of about 100,000 hours, if the high-temperature working stresses were restricted to about 7500 pounds per square inch. To a considerable degree this metallurgical restriction resulted in a heavier and bulkier design than would be necessary with the alloys available today. However, the 1942 premises have resulted in a power plant having about 29 per cent efficiency at full load, with a weight of about 30 pounds per horsepower and occupying a space of approximately 5 cubic feet per horsepower. In appraising this development it should not be forgotten that this unit is now three years old, and in view of the rapid advances in the field of gas turbines, it is in a sense already outmoded. Figure 1 shows a view of the plant on test at Jeannette, Pa.

Arrangement of the Plant.

The cycle 2-1-0.75, as described in (1), was selected to meet the technical requirements. Figure 2 shows a schematic arrangement of the unit which embodies this cycle, consisting of a low-pressure compressor *A*, driven by a free-floating high-pressure turbine *F;* and a high-pressure compressor *C*, driven by the low-pressure turbine *H*, from which the useful power is also extracted. After passing through the low-pressure compressor, the air is cooled in the intercooler *B*, recompressed in the high-pressure compressor and preheated in the regenerator *D*, from which it passes to the high-pressure combustion chamber *E* and is delivered to the high-pressure turbine. The exhaust from the high-pressure turbine is reheated in the low-pressure combustion chamber *G*, after which it passes to the low-pressure turbine and thence exhausts to the stack by way of the hot side of the regenerator.

A motor *I*, connected to the low-pressure compressor through an overrunning clutch, is used for starting. Partial by-pass ducts *K* and *L*, with the valves *M* and *N*, serve to isolate the high-pressure turbine and low-pressure compressor during starting.

The turbines and compressors were designed for full-load shaft efficiencies of 87.5 per cent and 82.5 per cent, respectively. The maximum pressure ratio was set at 6.5 and the temperature at the turbine inlets was limited to about 1200 degrees F. The regenerator was designed for 75 per cent air side effectiveness*. This cycle is identical with Figure 1, and its performance is predicted in Figures 6 through 10, of (1).

The gas turbine confronts the designer with much more rigid restrictions, particularly in regard to the accurate prediction of the performance of the various elements in the cycle, than does any other prime mover. Not only must the losses be controlled, but the pressure-flow characteristics of the equipment must be accurately predicted over a wide operating range in order to assure that the machinery elements are effectively matched at all loads. The selection of a positive-displacement compressor, which is inherently stable under all conditions, to a degree relieved the severity of these restrictions and at the same time made it possible to approach the ideal of variable-flow, constant-temperature operation at all loads. This ideal is not only thermodynamically attractive, but it also simplifies the problems of mechanical design through reduction of temperature gradients, a factor of particular importance in a naval plant, where a premium is placed on rapid maneuverability without sacrifice of reliability.

* This term was previously denoted by regenerator efficiency.

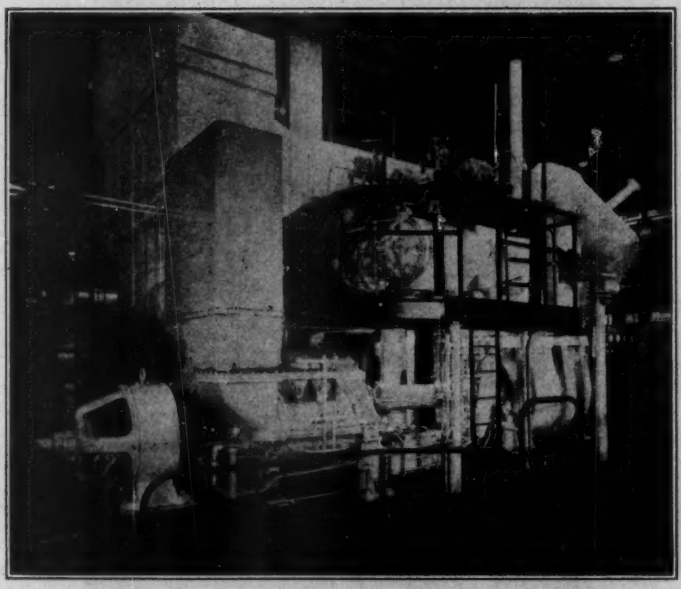


FIGURE 1.—2500-HORSEPOWER ELLIOTT-LYSHOLM GAS TURBINE PLANT.

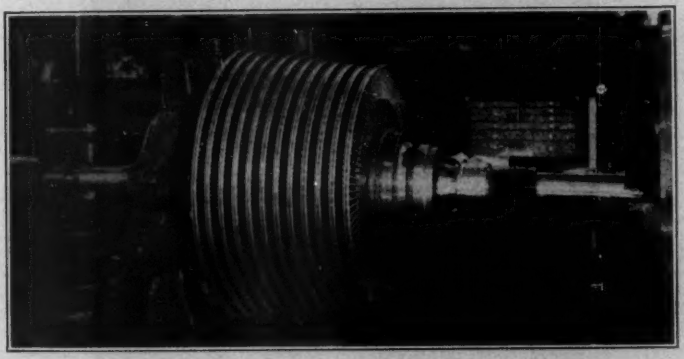


FIGURE 4.—HIGH-PRESSURE TURBINE ROTOR.

FIGURE 5.—HIGH-PRESSURE TURBINE DURING ASSEMBLY.

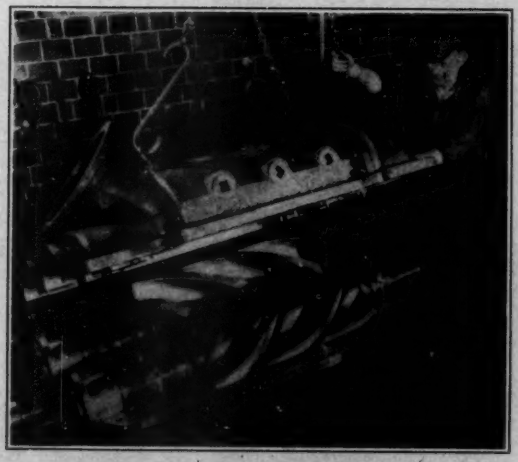


FIGURE 6.—HIGH-PRESSURE COMPRESSOR DURING ASSEMBLY.

FIGURE 11.—CONTROL PANEL.

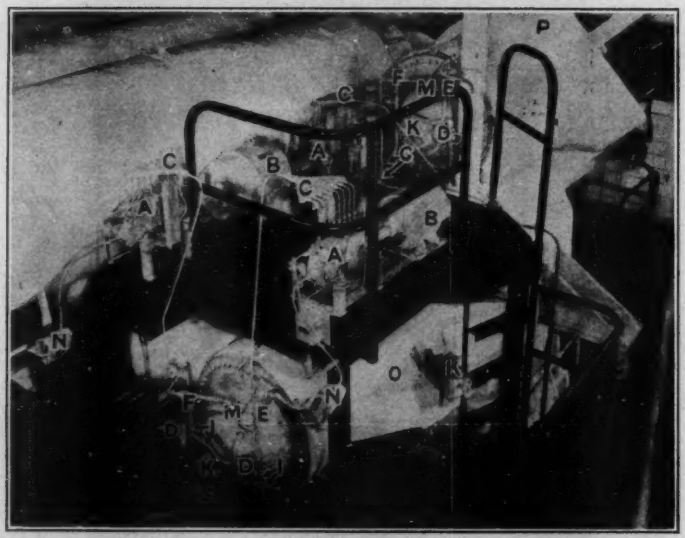


FIG 12.—VIEW OF COMBUSTION PLATFORM

A —Injection pumps
B —Injection pump motors
C —High-pressure fuel-oil supply headers
D—Injectors
E —Flame rod bodies
F —Pilot burner aspirator
G —Pressure control valves
I —Cooling oil lines to injectors
J —Ignition air and flame rod cooling air
K—Sight glass
M—Spark plug
N—Tachometers
O —High-pressure combustion chamber
P —Low-pressure combustion chamber

Constant-temperature operation is very nearly achieved in the present arrangement by driving the low-pressure compressor from a free-floating turbine, the speed of which is controlled by the fuel supplied to the high-pressure combustion chamber. The quantity of air circulated is roughly proportional to the speed of this combination. It might appear that either the high-pressure or the low-pressure turbine could be free floating, but a further consideration of the pressure-flow characteristics at light loads, and at starting, dictates the choice of the high-pressure turbine for this purpose, because its operating pressure ratio is nearly constant over a wide air flow range, inasmuch as the exhaust pressure, as well as the inlet pressure, of the high-pressure turbine is proportional to the flow. This is not the case for the low-pressure turbine, since its exhaust pressure is fixed. The energy available is proportional to the inlet temperature and the pressure ratio, with the result that, if the low-pressure turbine were isolated, its pressure ratio under the low flow conditions existing at light loads would be so reduced that its inlet temperature would be excessive.

Turbines.

The low and high-pressure turbines are of similar construction and are illustrated by the cross-section of the high-pressure machine, Figure 3. Details during manufacture are shown in Figures 4 and 5.

The turbines are multi-stage reaction machines, designed for approximately constant velocity ratio through each stage, with identical blades of constant length mounted on a tapered flow path. Since present metallurgical technique precludes the use of large forgings of certain heat-resisting alloys, the hollow rotor is fabricated from cross-rolled plates, with forged stub shafts at each end. The stationary blading is mounted on split diaphragms, supported by the inner casing.

The turbines are designed on a heat-elastic basis, whereby the various component parts, as well as the main structure, are supported on radial pins in such a fashion that correct centering is maintained while complete freedom for uniform radial expansion is allowed. Several of the radial pins supporting the labyrinth glands and diaphragms are shown in Figure 3. The main supporting pins at the exhaust and inlet ends also serve as heat-dams, across which a high-temperature gradient is maintained with the aid of cooling air introduced in the pin ring member. The main stationary parts are symmetrical castings, in which the metal masses are held to a minimum. An outer casing, for sealing the unit, is fabricated from thin sections, so that it is sufficiently flexible to absorb the piping strains without transmitting large forces to the main structure of the turbines. The outer casing can be telescoped into the exhaust ring to permit access to the inner casing. Each of the turbines is mounted rigidly on the bearing casings, one of which is fixed to the foundation and the other supported on links in order to provide for axial expansions of the order of ½ inch. Each turbine is provided with an electrically driven turning gear.

Compressors.

The Lysholm compressor is essentially a screw pump with the rotor so designed that the air is compressed within the casing. The low-pressure compressor is a double unit rated at approximately 25,000 cubic feet per minute at a pressure ratio of 3, while the high-pressure compressor is a single unit with one-half this displacement. Identical rotor pairs are employed in all compressors. Clearance between a pair of compressor rotors is maintained by timing gears with the result that the engaging surfaces of the compressor require no lubrication. Figure 6 shows an assembly view in the shop, while in Figure 7 is shown a section through the high-pressure compressor.

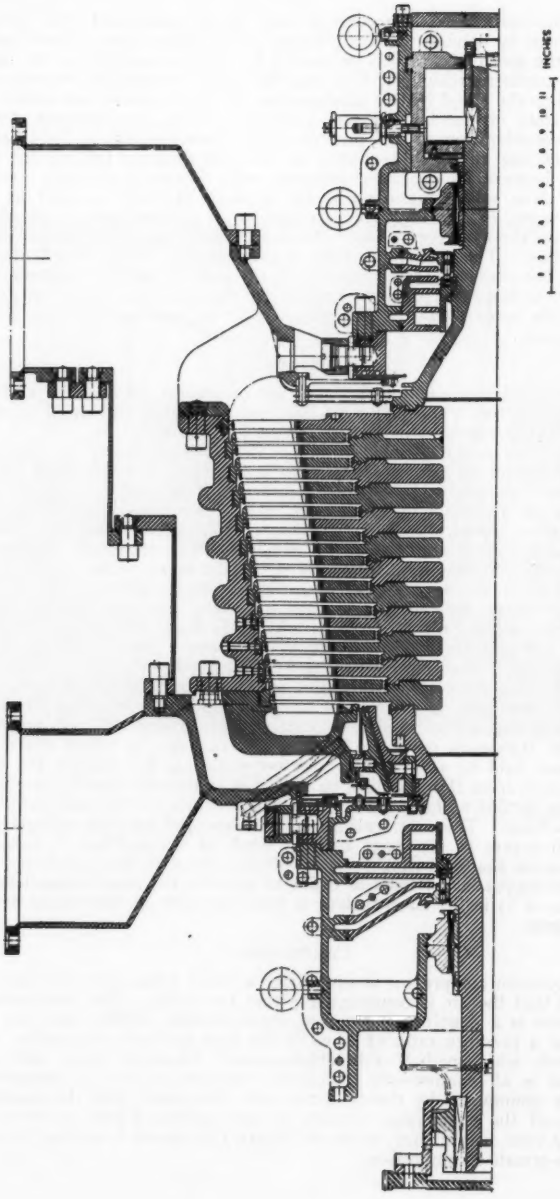


FIG. 3.—SECTIONAL VIEW OF THE HIGH-PRESSURE TURBINE

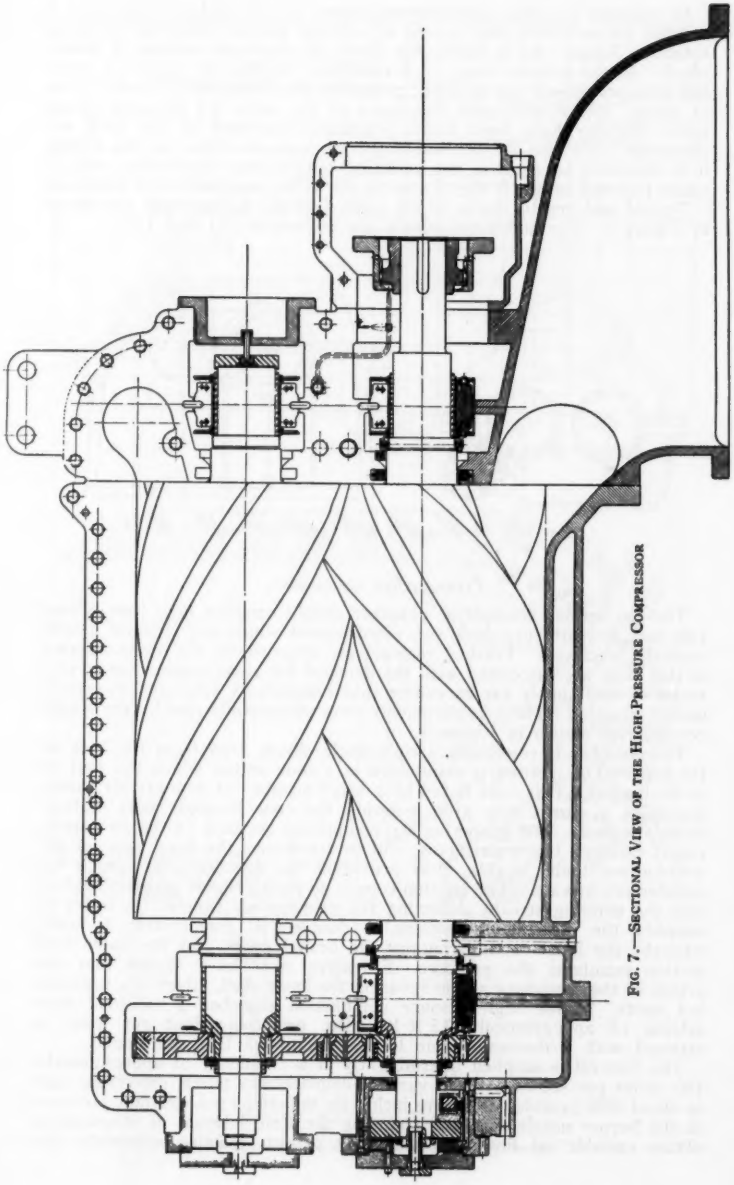


FIG. 7.—SECTIONAL VIEW OF THE HIGH-PRESSURE COMPRESSOR

In contrast to other displacement types, the Lysholm compressor is suitable for relatively high speeds of rotation without incurring excessive induction losses. As a result, the effect of clearance leakage is diminished. At the present stage of development, peripheral speeds of about 300 feet per second are possible, providing the compression ratio is two or more. Radial and axial clearances of the order 0.1 per cent of the rotor diameter have been found practical. Inasmuch as the inlet and discharge ports are located on diagonally opposite sides of the casing, it is desirable to equalize the resulting temperature distribution with a water-jacketed casing, if the clearances are to be maintained at a minimum.

Typical test results for a 12,500 cubic feet per minute unit are shown in Figure 8. For additional details see references (2) and (3).

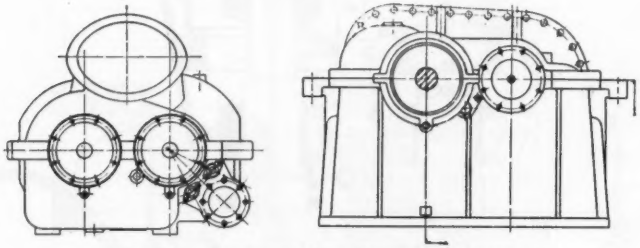


FIG. 7 (a).—END VIEWS OF HIGH-PRESSURE COMPRESSOR

COMBUSTION CHAMBERS.

The gas turbine combustion chamber should combine high heat release rate with low-pressure drop, two requirements which are, to some extent, mutually exclusive. Further restrictions, imposed by the cycle selected in this case, are associated with the demand for clean combustion over a range of fuel supply far exceeding that successfully employed in present marine practice. These requirements were successfully met by the simple construction shown in Figure 9.

The chamber is essentially a right-angle elbow, fired from the back of the bend. The ignition is maintained in a cone within which the fuel oil is discharged. This cone is fed by a small quantity of primary air, which maintains a vortex ring after entering the cone through slots midway along the cone. The tempering air, comprising the bulk of the air supply, enters through the main elbow, and in traversing the bend sets up the well-known double vortex, thus providing the necessary turbulence for satisfactory mixing. The ignition cone is so placed that it projects slightly into the entering stream, deflecting the primary air supply. In order to simplify the mechanical problem, a double-wall construction is used, whereby the inner section contains the heated gases, and the outer cold section maintains the pressure. Extensive tests have shown that the action of the secondary air, in scouring the inner shell, effectively prevents hot spots. In the high-pressure combustion chamber a full-load heat release of approximately 2.5×10^6 Btu. per cubic foot per hour is attained with a thermodynamic loss of about 0.5 Btu. per pound.

The fuel oil is supplied intermittently at a frequency of approximately 160 cycles per second from a variable-displacement pump, delivering fuel at about 3500 pounds per square inch. By maintaining a constant pressure at the burner nozzle tips, and changing the time interval of injection to obtain variable oil flow, the atomizing pattern remains essentially the

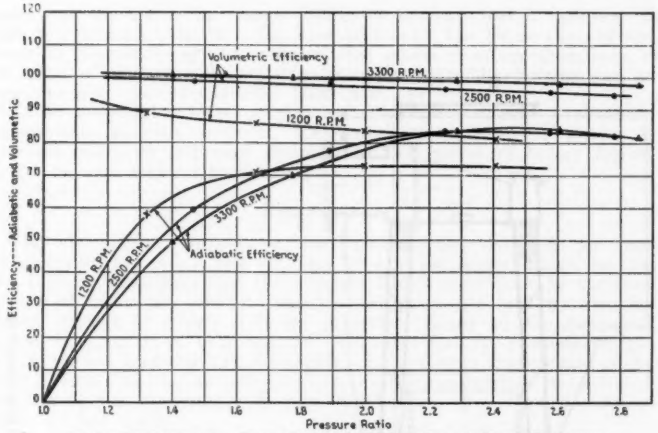


FIG. 8.—TEST RESULTS OF AN ELLIOTT-LYSHOLM 12,500 CUBIC FEET PER MINUTE COMPRESSOR

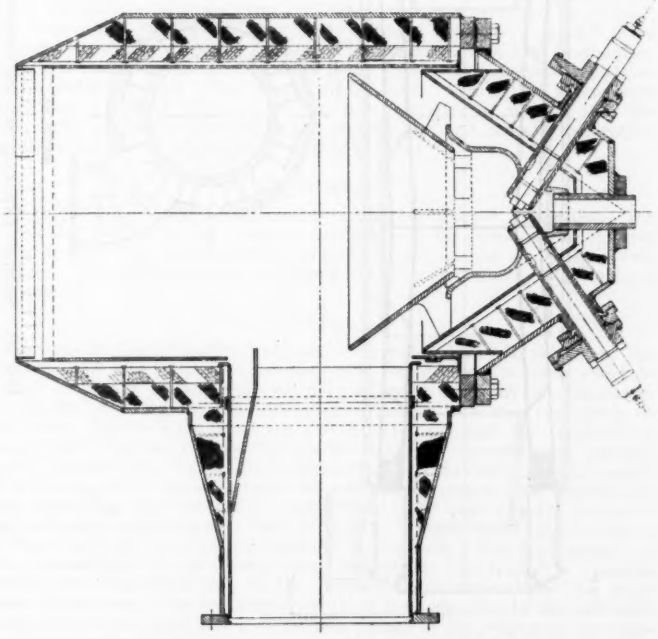


FIG. 9.—SECTION THROUGH THE ELLIOTT COMBUSTION CHAMBER

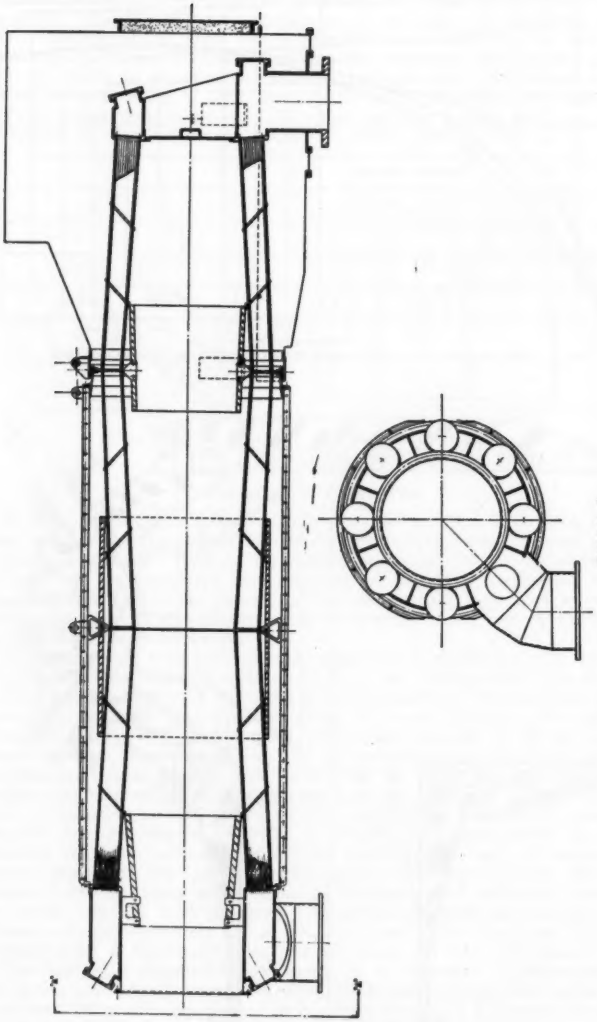


FIG. 10.—SECTION THROUGH THE AIR PREHEATER REGENERATOR

same over the entire load range. Clean combustion over a variation of fuel flow of more than 30: 1 has been achieved. The burners are lighted from an electrically ignited propane pilot, which is in operation only at the time of starting. By suitable bayonette locks the burners can be removed and replaced while the plant is in operation.

So far the unit has been operated with distillate fuel oils. Preliminary experiments indicate, however, that operation with lower grades is practical, although additional developments are necessary before bunker C can be used without trouble.

Regenerator.

The gas turbine regenerator, particularly when designed for high effectiveness, presents a new problem of heat exchanger design, in that the thermal resistances and the pressure losses must both be carefully controlled under the difficult conditions of gas-to-gas heat transfer between streams at widely different pressures. Many of the commonly known and attractive heat transfer structures lead to physical designs which are not geometrically compatible with the other components of the plant. When this plant was designed, the tubular construction was selected, because of the inadequacy of heat transfer and flow friction data on other surface types.

The first approach to the design of a 75 per cent effective regenerator, having a transfer area of approximately 3 square feet per horsepower was made by the Air Preheater Corporation and is illustrated in Figure 10, which shows one of three identical units which are arranged in parallel. The unit is of the tubular type with a single cold air pass and three diagonal counterflow hot gas passes. The tubes are 5/16 inch in diameter and are arranged in sections 14 rows deep. Means for thermal expansion are provided through a low-pressure slip joint. While this regenerator represents a distinct advance over the previous state of the art, it is the most bulky piece of equipment in the whole plant, and considerable future development can be anticipated on this element.

Intercooler.

With the ample water facilities available for marine applications, the intercooler design is not a major problem. A conventional automotive radiator type with oblong tubes, suitably finned on the air side, is employed. The cooling-water consumption is approximately 250 gallons per minute.

Control.

The gas turbine is adapted to simple control means. In this plant the controls and supervisory instruments are located on the main panel as shown in Figure 11, and the output is adjusted solely by the regulation of the fuel delivery through the three actuating levers in the center of the panel. On the main panel are also mounted pressure and temperature gages, speed indicators, starting control and the miscellaneous warning lights, actuated by the safety mechanisms. An auxiliary panel, located directly across the cat-walk from the main panel, contains the instruments for controlling the cooling-air supply and, for experimental purposes, an automatic potentiometer for recording critical metal temperatures.

Figure 12 is a view of the combustion platform. The fuel pumps, of the conventional Bosch Diesel type, are driven electrically at constant speed. The fuel delivered to the burners is determined by the relief ports in the pump cylinder, whose positions are controlled by the hydraulic actuators on the main panel. The two pumps in the high-pressure combustion chamber and the single pumps in the low-pressure chamber are separately actuated.

While the operation of this plant is essentially based on constant temperature, the desired load changes are brought about by transient temperature changes, largely in the high-pressure combustion chamber. If, when operating at any one load, it is desired to increase the plant output, the fuel injection rate to the high-pressure combustion chamber is increased. This causes a momentary increase in temperature, depending on the magnitude of the change in fuel setting, but generally less than 100 degrees F. As a consequence, the high-pressure turbine accelerates, and in so doing delivers an increased quantity of air to the system, which in turn increases the output and restores the temperature. A new equilibrium condition, therefore, is established within a few seconds. During this accelerating transient it is necessary, of course, to alter also the fuel rate to the low-pressure combustion chamber.

Under this type of control the plant can be maneuvered as rapidly as the load can be followed with the water brake. Maneuvers from very light load to full load in about 15 seconds have been made without trouble. Maneuvers from a full ahead speed of 3000 revolutions per minute to a no load speed of about 1400 revolutions per minute, and thence back to 3000 revolutions per minute, have been completed in 90 seconds without any signs of distress. In these maneuvers the water brake setting remained constant, and they therefore represent the most severe conditions that can be imposed in land trials without excessive complications.

The plant is started by means of a 50-horsepower direct-current motor, with the high-pressure turbine and low-pressure compressor isolated from the remaining equipment by the partial by-pass valves shown in Figure 2. The by-pass valves as well as the starting motor are electrically controlled by the operator from the main panel. The combustion chambers are ignited from the pilot burner, whose control is also centered at this point. From a cold start the plant is generally warmed up by using one burner in the high-pressure combustion chamber, while the starting motor is held at a speed of about 900 revolutions per minute until the plant is hot. During this period the low-pressure turbine and the high-pressure compressor accelerate from the turning-gear speed of about 10 revolutions per minute to about 300 revolutions per minute, heating these elements as well. The gas temperature is gradually brought to about 1000 degrees F., with the rate of temperature increase established by the relative expansion between the stationary and the rotating turbine parts.

The two independently running units are now coupled by first accelerating the low-pressure turbine by partially closing its by-pass valve, and then increasing the starting motor speed until the flow through the high-pressure compressor by-pass is negligible. At this point both by-passes are closed and the starting-motor speed increased, together with the high-pressure fuel-injection rate, until the plant is self-sustaining. The starting motor, connected by an over-running clutch, automatically disengages itself. Additional load is obtained by igniting the second high-pressure burner from the flame already existing in that chamber, and by igniting the low-pressure burner by means of its propane pilot flame. Once the warm-up period is complete, the plant is ready to deliver any power within its design capacity.

The safety alarms, also on the main control panel, consist of the usual over-temperature and under-pressure lubricating-oil signals, and a Wheelco Flame-o-trol unit for each combustion chamber, to give a visual signal in case the flame is blown out, and, at the same instant to cut off automatically further fuel injection. As an added precaution, so far not needed, explosion diaphragms are located in the duct work. All the auxiliaries in the plant are electrically driven, but in case of failure of electrical power a conventional steam-driven stand-by oil pump is available.

TESTS.

Prior to the assembly of the plant, each of the machinery components was subjected to extensive tests. These preliminary tests, while not reported here, consumed a considerable portion of the development efforts, but contributed greatly to its success. It is our conviction that without a procedure of this kind a major machinery development is well-nigh hopeless.

The plant was first operated as a unit on October 28, 1944, and following a period in which minor difficulties were corrected the test program began in December, and the acceptance tests were made on December 28, 1944. Since that time the unit has remained at the Jeannette plant for more detailed investigations in order to establish a background for developments in the future. The unit has operated a total of nearly 500 hours under load, including one continuous period of 48 hours.

A number of performance trials have been made, two of which are summarized in Tables 1 and 2. Test run No. 9, given in Table 1, constituted the acceptance test. Alterations for purposes of investigation are being made continually, and no two test runs are identical. Run No. 16, given in Table 2, for example, differs from run No. 9 in that greater losses were deliberately introduced at the intake filter and at the high-pressure combustion chamber inlet. Moreover, fouling of the intercooler with sound-insulating material occurred during this test.

With present-day technique, accurate evaluation of the performance of the individual components, when assembled in a gas turbine plant, is not practical. The reliable measurements are those of power and fuel flow, significant primarily from the point of view of overall plant performance. In the test data presented it is believed also that the pressure measurements are reasonably accurate, inasmuch as multiple readings were taken, in many instances as direct pressure differences. However, the accuracy of the high-temperature observations is questionable, for even with the specially developed, seven shielded temperature probes, stratification and radiation introduce significant errors. Fortunately, many of these measuring difficulties can be eliminated on single-element tests. For example, in the tests of the high-pressure turbine alone, entirely satisfactory agreement between state measurements and power tests was obtained, thus further emphasizing the need for thorough component testing.

The performance data given in Tables 1 and 2 are direct readings without any corrections other than instrument calibration. To put this performance on a rational basis, it is necessary to correct to a standard 70-degree F. inlet temperature, to eliminate intake filter and stack losses not properly a part of the unit, and to correlate the two test runs with respect to variations in intercooler and combustion chamber losses.

For this purpose a study, the basis for which is explained more fully in the Appendix, has been made on the effect of the various parastic losses on the available power delivered to the propeller shaft. These losses, summarized in Figure 13, are identical for both test runs, except in respect to the high-pressure combustion chamber and the intercooler, which are at variance for the reasons previously explained. The performance of run No. 16 was then estimated with the same combustion chamber and intercooler losses as for run No. 9.

To these results were applied additional corrections for the power consumed by the following auxiliaries, driven from an external power source: cooling air, transfer and main fuel pumps, lubricating-oil pumps, and water circulating pumps. Every correction applied to the basic data of Tables 1 and 2 is shown in Table 3.

As a matter of interest, the parasitic losses at full load, shown in Figure 13, have been converted to Btu per pound and are compared in

TABLE 1.—DATA FOR TEST RUN 9, DECEMBER 28, 1944

Test No.	9A	9B	9C	9D
Barometric pressure, psi abs	14.385	14.382	14.378	14.384
Inlet pressure, low-pressure compressor, psi abs	14.28	14.18	14.03	13.88
Discharge pressure, low-pressure compressor, psi abs	29.25	33.71	....	35.43
Inlet pressure, high-pressure compressor, psi abs	29.21	33.64	....	35.16
Discharge pressure, high-pressure compressor, psi abs	40.63	54.33	69.58	83.48
Inlet pressure, high-pressure combustion chamber, psi abs	40.08	53.42	68.44	82.12
Inlet pressure, high-pressure turbine, psi abs	39.86	53.13	68.08	81.68
Discharge pressure, high-pressure turbine, psi abs	25.27	31.55	39.13	46.74
Inlet pressure, low-pressure turbine, psi abs	25.14	31.37	38.88	46.43
Discharge pressure, low-pressure turbine, psi abs	14.39	14.45	14.50	14.55
Discharge pressure, regenerator, psi abs	14.38	14.40	14.43	14.45
Inlet temperature, low-pressure compressor, deg F	77.5	78.6	79.4	72.6
Discharge temperature, low-pressure compressor, deg F	233.0	270.0	288.0	286.0
Inlet temperature, high-pressure compressor, deg F	63.6	67.3	77.1	85.1
Discharge temperature, high-pressure compressor, deg F	143.8	176.0	226.0	285.0
Discharge temperature, regenerator air side, deg F	876.0	798.0	735.0	727.0
Inlet temperature, regenerator gas side, deg F	1036.0	964.0	884.0	858.0
Discharge temperature, regenerator gas side, deg F	368.0	382.0	409.0	449.0
Inlet temperature, high-pressure turbine, deg F	1284.0	1255.0	1244.0	1251.0
Inlet temperature, low-pressure turbine, deg F	1240.0	1231.0	1210.0	1244.0
Speed of high-pressure turbine, rpm	1865.0	2465.0	3120.0	3720.0
Speed of low-pressure turbine, rpm	1660.0	1972.0	2403.0	3012.0
Oil flow, lb per hr	378.0	597.1	854.1	1112.6
Air flow, lb per sec	13.16	18.00	23.34	28.13
Brake horsepower	590.0	1108.0	1756.0	2335.0

TABLE 2.—DATA FOR TEST RUN 16, JANUARY 30, 1945

Test No.	16A	16B	16C	16D
Barometric pressure, psi abs	14.23	14.23	14.23	14.23
Inlet pressure, low-pressure compressor, psi abs	14.12	13.99	13.84	13.64
Discharge pressure, low-pressure compressor, psi abs	27.69	33.60	34.41	34.61
Inlet pressure, high-pressure compressor, psi abs	27.39	33.07	33.60	33.60
Discharge pressure, high-pressure compressor, psi abs	39.03	55.03	68.63	82.38
Inlet pressure, high-pressure combustion chamber, psi abs	38.35	54.04	67.55	81.15
Inlet pressure, high-pressure turbine, psi abs	37.99	53.53	66.93	80.43
Discharge pressure, high-pressure turbine, psi abs	24.36	31.82	38.60	45.66
Inlet pressure, low-pressure turbine, psi abs	24.23	31.63	38.33	45.33
Discharge pressure, low-pressure turbine, psi abs	14.26	14.30	14.34	14.40
Discharge pressure, regenerator, psi abs	14.24	14.25	14.27	14.30
Inlet temperature, low-pressure compressor, deg F	77.4	77.7	77.5	76.4
Discharge temperature, low-pressure compressor, deg F	222.4	268.7	280.9	290.5
Inlet temperature, high-pressure compressor, deg F	54.5	64.7	74.6	79.5
Discharge temperature, high-pressure compressor, deg F	135.4	176.7	227.9	282.3
Discharge temperature, regenerator air side, deg F	865.2	806.6	738.0	712.0
Inlet temperature, regenerator gas side, deg F	1023.0	972.0	880.0	841.0
Discharge temperature, regenerator gas side, deg F	361.0	389.0	413.0	443.0
Inlet temperature, high-pressure turbine, deg F	1243.0	1278.0	1236.0	1269.0
Inlet temperature, low-pressure turbine, deg F	1225.0	1253.0	1209.0	1224.0
Speed of high-pressure turbine, rpm	1804.0	2496.0	3111.0	3770.0
Speed of low-pressure turbine, rpm	1662.0	1993.0	2460.0	2986.0
Oil flow, lb per hr	374.0	627.0	849.0	1074.0
Air flow lb per sec	12.69	18.24	23.24	28.15
Brake horsepower	494.0	1148.0	1674.0	2148.0

TABLE 3.—CORRECTED TEST PERFORMANCE

Test No.	9A	9B	9C	9D	16A	16B	16C	16D
Tested brake horsepower	590.0	1108.0	1756.0	2335.0	494.0	1148.0	1674.0	2148.0
Tested fuel flow, lb per hr	378.0	597.1	854.1	1112.6	374.0	627.0	849.0	1074.0
Tested fuel rate, lb per hp-hr	0.641	0.539	0.486	0.476	0.757	0.546	0.507	0.500
Per cent power corrections								
Inlet temperature, 70 deg F	1.93	1.56	1.61	0.44	1.99	1.52	1.33	1.15
Inlet duct and filter	1.57	1.81	2.80	3.93	1.84	2.35	3.34	5.08
Stack	0.22	0.25	0.48	0.60	0.22	0.25	0.41	0.65
High-pressure combustion chamber	0	0	0	0	1.05	0.59	0.49	0.47
Air side intercooler	0	0	0	0	2.26	1.93	2.57	2.86
Total	+3.72	+3.62	+4.89	+4.97	+7.36	+6.64	+8.14	+10.21
Gross corrected, bhp	612.0	1148.0	1842.0	2451.0	530.0	1224.0	1810.0	2367.0
Per cent efficiency corrections								
Inlet temperature, 70 deg F	−1.86	−1.48	−1.51	−0.41	−1.93	−1.44	−1.25	−1.07
Inlet duct and filter	−1.51	−1.71	−2.62	−3.67	−1.78	−2.23	−3.14	−4.76
Stack	−0.18	−0.20	−0.38	−0.47	−0.18	−0.20	−0.32	−0.52
High-pressure combustion chamber	0	0	0	0	−1.02	−0.56	−0.46	−0.44
Air side intercooler	0	0	0	0	−2.19	−1.83	−2.43	−2.69
Total	−3.55	−3.39	−4.51	−4.55	−7.10	−6.26	−7.60	−9.48
Gross corrected fuel rate	0.618	0.521	0.465	0.456	0.707	0.515	0.471	0.457
Gross corrected fuel flow, lb per hp-hr	378.2	598.1	856.5	1117.6	374.7	629.7	852.5	1081.0
Auxiliary power loss								
Injection pumps, hp	4.4	4.8	5.7	7.4	4.3	4.6	5.3	6.3
Cooling air, hp	35.7	43.2	51.6	56.3	35.7	43.2	51.6	56.3
Lube oil pumps, hp	8.2	8.2	8.2	8.2	8.2	8.2	8.2	8.2
Circulating water pumps, hp	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
Total auxiliary power, hp	48.8	56.7	66.0	72.4	48.7	56.5	65.6	71.3
Net corrected bhp	563.2	1091.3	1776.0	2378.6	481.3	1167.5	1744.4	2295.7
Net corrected fuel rate, lb per hp-hr	0.672	0.548	0.482	0.470	0.779	0.539	0.489	0.471
Thermal efficiency (LHV), per cent	20.41	25.04	28.11	29.22	17.62	25.38	27.70	29.43

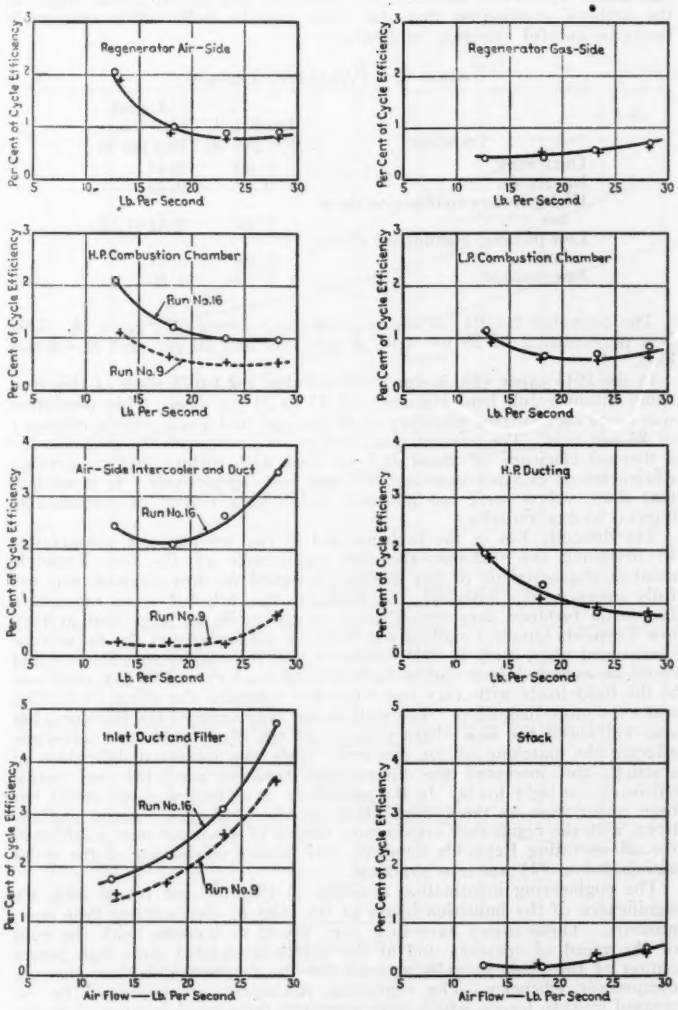


FIG. 13.—PARASITIC LOSSES OF THE PLANT

Thermal efficiency (LHV), per cent	20.41	25.04	28.11	29.22	17.62	25.38	27.70	29.43

Table 4 with the tentative suggestions presented in reference (1). At that time these data were generally viewed as overly optimistic, but the substantial agreement between the predicted and actual losses supports the authors' convictions that the losses can be held within reasonable limits by careful attention to detail.

TABLE 4.—PARASITIC LOSSES

Location	Predicted Btu per lb	Actual (4/4 load), Btu per lb
Duct work	0.80	0.44
Intercooler	0.37	0.45
High-pressure combustion chamber	0.54	0.31–0.52
Low-pressure combustion chamber	0.54	0.41
Regenerator	1.10	0.86

The corrected results for the two runs are shown on Figure 14. This is a performance of 29 per cent at full load and 20 per cent at quarter load.

In the 1943 paper (1), it was predicted that a 2-1-0.75 plant of this type could attain a full load efficiency of 32 to 33 per cent. This prediction was based on a turbine efficiency of 89 per cent and a compressor efficiency of 83 per cent. The present plant was more conservatively designed for a thermal efficiency of about 31.5 per cent with turbine and compressor efficiencies of 87.5 per cent and 82.5 per cent, respectively. It is evident that these values were not attained, and it is a matter of considerable interest to explain why.

The difficulty lies in the turbines and in the low-pressure compressor. In designing the turbines, the full significance of the low Reynolds number, characteristic of gas turbines designed for low stresses, was not fully appreciated. Although the blade pitches selected were reasonable for steam turbines, they nevertheless proved to be so large that at very low Reynolds numbers stalling was induced, particularly at the tip section. Component tests, early in 1944, indicated that the maximum efficiency that could be expected from either turbine would not exceed 85 per cent, and at the light loads with very low Reynolds numbers, the effect of stalling was even more magnified. The stalling not only reduced the efficiency, but also influenced the flow characteristics of the blade path, and adversely affected the matching of the elements. This also increased difficulties of starting, and increased the discrepancy between predicted and actual efficiencies at light loads. In the meantime, an extended development has been undertaken on the turbine blade problem for these special applications, with the result that satisfactory ranges of pitch are now established for all operating Reynolds numbers, and turbine efficiencies of the order anticipated in (1) are now practical.

The engineering information available in 1943 did not reveal fully the significance of the induction losses at the inlet to displacement type compressors. These losses have now been found to increase with the cube of the speed of rotation, and at the speeds associated with high power output of the plant partially account for the discrepancy in low-pressure compressor efficiency. The remaining discrepancy is explained by increased leakage losses, which were purposely introduced because of manufacturing difficulties with the rotor profiles. By a more informed choice of rotating speed and proper tooling, both of these difficulties have been resolved, and compressors of later construction have met the original expectations.

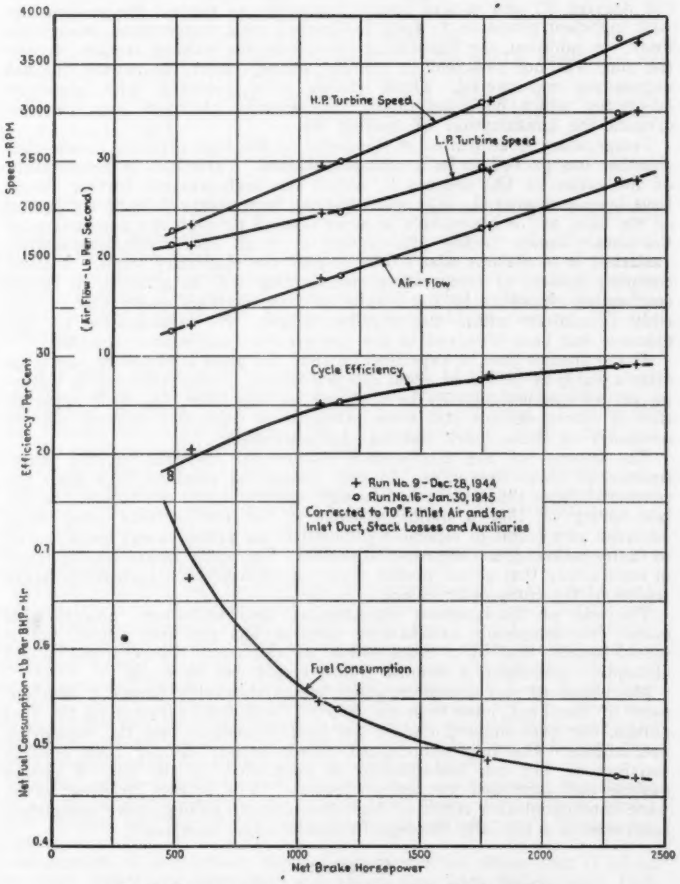


FIG. 14.—PERFORMANCE OF THE ELLIOTT 2500-HORSEPOWER GAS TURBINE PLANT

Troubles.

Apart from the situation brought about by the failure to appraise correctly the influence of Reynolds number the design of the turbines has proven satisfactory, with the exception of the carbon glands in the high-pressure machine. Although a special binder was employed in the manufacture of the carbon rings, they disintegrated at temperatures above 850 degrees F., and it was found impossible to control the cooling air with sufficient precision to keep the carbon ring temperature below this limit. In addition, the lubrication offered by the leakage stream beneath the ring was not sufficient to prevent seizing under the severe thermal expansions encountered. These glands were replaced with step-type labyrinths, which have functioned satisfactorily, although they have increased the consumption of cooling air.

Temperature stratification at the outlet of the high-pressure combustion chamber has proved to be a serious problem. Variations in temperature of the order of 150 degrees F. across the high-pressure turbine flange have been encountered. The variation has been observed to be a function of the load, and is susceptible to some control through the positioning of the burner cones. While 150 degrees F. is an appreciable temperature variation, it is obvious that it affects only the stationary parts. A more complete control of temperature stratification will be achieved in future combustion chambers by the results of investigations under way on the eddy formations within the chamber proper. No stratification of significance has been observed in the low-pressure combustion chamber.

At the present time, in experimental runs, the plant is placed in operation after a warm-up period of about 2½ to 3 hours. Some reduction in warm-up period can undoubtedly be obtained on this plant, but it is apparent that in future designs still more attention to thermal expansion will be necessary to attain quick starting characteristics.

The success of the displacement compressor depends on the maintenance of close clearances. In this connection troubles have been encountered from the entrance of foreign materials into the rotors. In the late spring of 1945 a major casualty in the low-pressure compressor occurred as a result of clearance difficulties, the prime causes being traced to faulty jacketing and improper alinement. The water jacket was designed in such a way that an air pocket occurred adjacent to a high-temperature region of the compressor casing.

The use of displacement compressors also introduces objectionable noise. No completely satisfactory solution has yet been found, but it would appear that by a combination of resonance dampers and sound absorption treatment, a definite improvement can be made.

Miscellaneous mechanical troubles, which inevitably plague a development of this kind, have been encountered with the operation of the fuel pumps, the over-running clutch, the heat insulation, and the measuring instruments. The high-temperature flange design has not been entirely satisfactory, but can undoubtedly be controlled by the use of proper gaskets and improved mechanical features. The bolting troubles, which were anticipated as a result of high temperature galling, were completely overcome by a specially developed colloidal silver compound.

Conclusion.

This development is only the beginning; there are a great many problems to be solved before the gas turbine becomes an established prime mover. With the confidence gained through the successful operation of this plant, there is every reason to expect that these problems will be attacked with vigor and intelligence. The incentive should be even greater, for recent metallurgical developments open new possibilities for advances in the operating temperature, with resulting improvements in

efficiency and reduction in size. While far from perfect, the Elliott gas turbine has given encouraging results, and it has paved the way for the three additional units now under construction.

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MODERN SHIPS OF THE AMERICAN MERCHANT MARINE.

This article, discussing the requirements of vessels for effective participation in maritime trade and covering many of the advances of the last quarter century in marine power plant design, is reproduced from the General Electric Review of December, 1945. The author is Frank V. Smith of the Federal and Marine Divisions of the General Electric Company.

The United States now has the largest fleet of merchant ships ever assembled under one flag. This fleet consists of approximately 5300 ships aggregating 54,000,000 tons as compared to a world total of 69,000,000 tons existing under all flags in 1939. The acquisition of this fleet has cost the taxpayer roughly $15,000,000,000.

Prior to World War I, less than nine per cent of our foreign commerce was carried in American ships. In World War I, however, a bridge-of-ships plan was inaugurated and upwards of 4,000,000 tons of ships were built in a single year—a world record at the time. At the end of this first war we had acquired a fleet of merchant ships having a total tonnage of 17,000,000 tons.

Between 1920 and 1939 the tonnage of merchant ships flying the American flag fell from 17,000,000 tons to 11,470,000 tons. Only 2,700,000 tons of this fleet was engaged in the foreign trade lanes. Of the rest, 6,100,000 tons was engaged in the Great Lakes and coastal trade routes, and the remainder of the fleet was idle. During this period, the percentage of our imports and exports carried in American ships fell from 50 per cent to 25 per cent. During this period of retrogression of the American Merchant Marine, the Axis Powers—Germany, Japan, and Italy—increased their fleet of merchant ships by 7,168,000 tons. American patronage was largely instrumental in the building up of these fleets.

Already, people not conversant with the facts are beginning to question our ability to operate ships in the foreign trade lanes, our ability to render equal service, and our ability to compete successfully with foreign-flag ships.

An analysis of our present fleet of merchant ships shows that if all of the ships which are outmoded and all of the ships which were built primarily as a war emergency measure are scrapped, there will still remain a formidable fleet of 2200 ships with an aggregate tonnage of 24,000,000 tons. This latter fleet consists of combination freight and passenger ships, cargo ships, and oil tankers. These ships are highly efficient and sufficient in number to carry our rightful share of 50 per cent of our imports and exports. Their continued use is justified both from the standpoint of national security and as a protection to our national economy.

These ships need no apology. All fields of research and the best engineering talent within our shores has had a hand in their making. Every effort has been made to combat the lower wage levels and lower building costs of other nations by superior engineering. This fleet of ships served well during the crisis and now forms the backbone of our postwar fleet.

The advances which have been incorporated in this postwar fleet are representative of the most modern trends in mechanical and electrical engineering practice and they find no counterpart in ships built in foreign countries prior to the war. Their sea speeds also excel those of ships of like class of all other maritime powers.

Engineering Advances.

It is interesting to note the magnitude of the engineering advances which have been made in marine power plant design during the past twenty-five years. Prior to World War I, the power plants on ships consisted primarily of coal-fired Scotch boilers, reciprocating engines, and steam-driven auxiliaries. Electric auxiliaries were not yet fully developed for such service and the only piece of electrical equipment on the majority of ships consisted of a small engine-driven generator for furnishing light.

Steam pressures in the majority of cases did not exceed 200 pounds and the application of turbines for propulsion was still awaiting a satisfactory gear design to effect an adequate speed reduction between a high-speed turbine and a low-speed efficient propeller. Even under these unfavorable circumstances, however, a bridge-of-ships program was launched and the turbine was forced into use as a war measure.

The turbine at this time had a good many hurdles to jump before it could come into its own. Steam auxiliaries which were applied on these ships were of the same type previously developed for use on reciprocating-engine drives and their high steam consumption in combination with the low steam consumption of the turbine created an excess of auxiliary exhaust steam which could not be entirely utilized for feed-water heating. This excess exhaust steam was by-passed to the main condensers which in turn reduced the vacuum and seriously impaired the turbine economy. The losses were pyramidial in effect as any attempt to speed up the condenser auxiliaries to improve the vacuum only tended to create a greater excess of auxiliary exhaust steam.

Studies of these early installations proved to be of great value to the future of marine engineering. It brought to light the fact that a marine power plant, including all of the auxiliary equipment, must work together as a closely integrated unit. Electric auxiliaries which derived their power from an auxiliary turbine-generator set operating condensing, became the answer to the excess auxiliary exhaust problem.

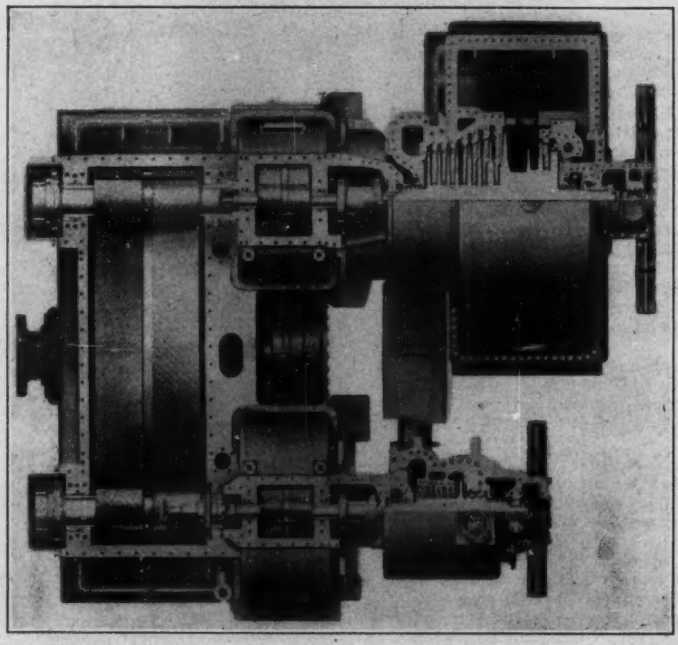
Lessons learned from the actual operation of the early gear designs, combined with intensive research, development, and improvement in machine-tool design also freed the turbine from its former restrictions and made possible the application of efficient high-speed turbines at low-output speeds at the propeller shaft.

When the turbine was divorced from the hindrances of the auxiliaries and gear limitations of the past, it became free to expand its field of application. High steam pressures and temperatures were adopted as well as the regenerative method of feed-water heating by the extraction method.

The boiler manufacturers were equally interested in improving the fuel economies on ships; and during this period of evolution new designs were developed to keep pace with the ever-increasing demand of the turbine manufacturers for higher pressures and temperatures. Automatic combustion control was also adopted to replace the haphazard practices of

GEARED STEAM TURBINE-GENERATOR SET FOR MARINE A-C AND D-C SERVICE.

THE U. S. S. "ADMIRAL W. S. BENSON" EQUIPPED WITH TURBINE-ELECTRIC PROPULSION EQUIPMENT, TWIN SCREW, NORMAL RATING 9000 HP., 120 RPM. PER SCREW.



CROSS-COMPOUND GEARED (SPEED-REDUCING) STEAM TURBINE UNIT FOR SHIP PROPULSION.

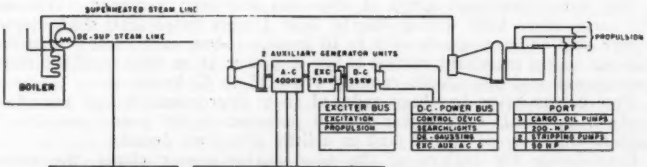


Fig. 1. Diagram of power sources in a modern turbine-electrically propelled tanker. Note the availability of power from the main propulsion plant for driving the cargo oil pumps

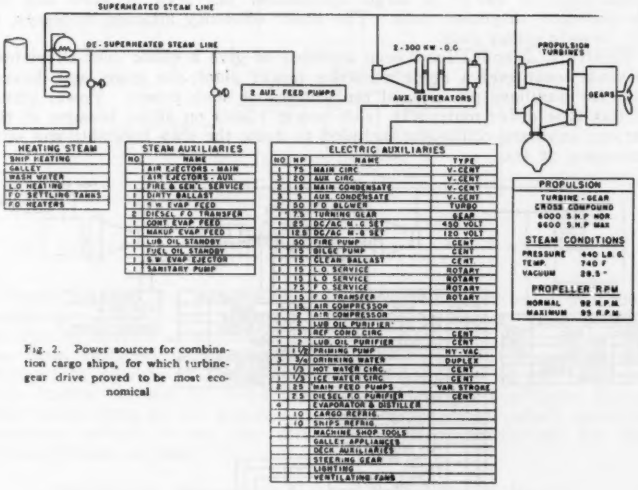


Fig. 2. Power sources for combination cargo ships, for which turbine-gear drive proved to be most economical

the past in which the matter of air regulation was left entirely to the fireroom personnel, who were seldom conversant with the fine art of burning fuel oil for the attainment of maximum efficiency.

Modern Marine Power Plants.

The size, power, and speed of ships has also been increased. Whereas the cargo ships built during World War I were rated 2500 shaft horsepower, and had sea speeds of 9 to 10 knots, modern cargo ships constituting our newer merchant marine range in power from 6000 to 12,800 shaft horsepower and sea speeds ranging from 15 to 17 knots.

Fuel consumption has been reduced from approximately one pound of fuel oil per shaft horsepower for all purposes to 0.6 pound and less, a gain in efficiency of 40 per cent in a little over two decades.

Considering the ratings of the new marine power plants, the initial steam conditions selected compare very favorably with the latest central-station practice on shore.

Marine power plants of 4000 to 12,800 Shp. have, in the majority of installations, adopted initial steam conditions of 450 pound gauge, 750 F., and in ratings of 18,000 Shp. and above, 600 pound gauge, 825 F. Four ships have adopted 1400 pound gauge pressure, 750 F., with two stages of reheat.

Full use has been made of extraction steam for feed-water heating and it has been the policy in the majority of cases to use three stages of such heating to give a final feed-water temperature of 300 F. In installations of this type both air preheaters and economizers have been installed, resulting in an over-all boiler efficiency of 88 per cent. In installations using but two stages of feed-water heating, resulting in a final feed-water temperature of 240 F., a larger economizer has been installed and the air preheater dispensed with. The boiler efficiency attained, however, is the same in either case.

Figures 1, 2, and 3 have been included to give a quick over-all picture of what constitutes a modern marine power plant, the many and diverse uses for auxiliary power, and the sources of such power. Power plants of this type differ materially from power plants on shore because of the various auxiliary equipment included to make the ship habitable and self-sustaining at sea.

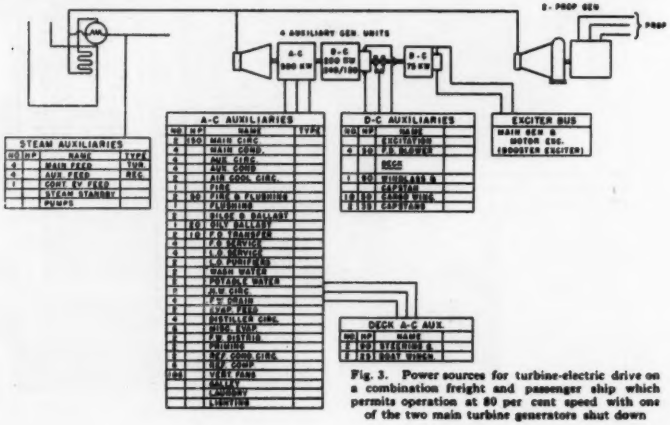


Fig. 3. Power sources for turbine-electric drive on a combination freight and passenger ship which permits operation at 80 per cent speed with one of the two main turbine generators shut down

Figure 1 represents the power sources in a modern turbine-electrically propelled tanker. This type of propulsion was selected because of the availability of power from the main propulsion plant in port for driving the cargo oil pumps. In the older types of installations it was not unusual for oil tankers to take from 24 to 36 hours to unload. In this newer type of installation 150,000 barrels of oil can be unloaded in 8 hours, resulting in a saving of one lay day in port.

Figure 2 shows the power source on a modern cargo ship. The cargo handling equipment on these ships consists of 16 high-speed cargo winches in which graduated speed control is afforded in the hoisting direction and graduated speed control with dynamic braking in the lowering direction. The capstans and anchor windlass are also of the motor-driven type. For the type of service, and in view of the fact that cargo ships operate on long voyages at constant speed, turbine-gear drive proved to be the most economical type of drive when all factors were considered.

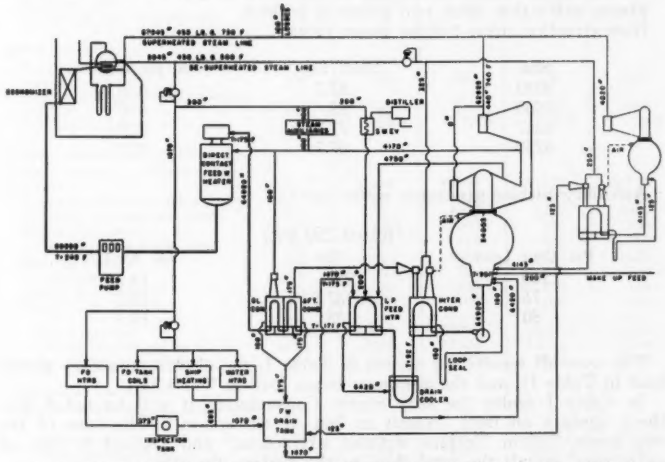


Fig. 4 Heat-balance diagram for 8500-shp cargo ship. Charts of this type aid the designer in co-ordinating the factors of design and in choosing the rating of the auxiliary equipment, size of piping, etc.

Figure 3 shows the power sources on a turbine-electrically propelled combination freight and passenger ship. Turbine-electric drive permits the ship to operate at approximately 80 per cent speed with one of the two main turbine-generators shut down. This proves economical for ships which operate at different speeds between ports of call on given trade routes. It will be noted that the engine-room auxiliaries derive their power from an a-c generator. The d-c generators which serve as generator exciters at sea also serve to furnish direct-current for the deck-auxiliaries in port.

Preliminary Studies of Marine Power Plant Design.

The over-all efficiency of a marine power plant is subject to calculation, and comparative studies are generally made of many different layouts before a specific selection is made. A typical thermodynamic analysis of an 8500-Shp. single-screw turbine-gear driven cargo ship is included in this article as a matter of interest. Studies of this type aid the designer

in co-ordinating the factors of design and in choosing the rating of the auxiliary equipment, size of piping, type of heat transfer apparatus, and ratings of boilers, condensers, and other miscellaneous apparatus. For clarity such studies are shown in pictorial form, as in Figure 4. Past experience has shown that the actual operating results check remarkably close with the calculated values.

The factors chosen as the basis for the typical design are as follows:
Boiler efficiency..... 88 per cent
Heat value of fuel oil..... 18,500 Btu. per pound
Steam temperature at turbine throttle..... 740 F.
Steam pressure at turbine throttle..... 440 pound gauge
Absolute pressure at turbine exhaust..... 1½ inch Hg.
Two stages of feed-water heating consisting of low-pressure heater for elevating temperature to 175 F. (at 8500 Shp.) and direct-contact de-aerating heater for raising temperature to 240 F.
Steam extraction from two points in turbine.
Nonextraction main turbine water rates:

Shp.	Shaft Rpm	Lb. per Shp.-Hr.
9350	87.7	6.97
8500	85.	6.97
6367	77.75	7.25
4250	67.7	7.75

Auxiliary turbine-generator water rates:

	(Rated 250 kw)	
Per Cent Load	Kw	Lb. Kw-Hr.
100	250	14.7
75	187.5	15.1
50	125.	16.3

The over-all results are shown in Table I, the electric auxiliary power load in Table II, and the auxiliary steam load in Table III.

In Table I under the title Steam Consumption it will be noted that there appears an item "Admit in lieu of extraction." The sum of the two items, "Main Turbine without extraction," and "Admit in lieu of extraction" equals the total flow to the turbine throttle.

The extraction water rates of marine turbines vary widely in marine installations. Extraction steam is used in many installations for the evaporators and for heating the vessel, which increases the demand for extraction steam, whereas the availability of steam auxiliary exhaust lessens the need for such extraction. These variations make it necessary to calculate each layout on its own merits. The final results of the study are shown in Figure 5.

Short Method of Evaluating Economic Measures.

Many times it is found desirable to evaluate the relative economics of specific features of design without undertaking the calculation of a complete heat balance. Such studies are of great value in projecting future engineering trends, and a great portion of the advances made to date may be accredited to studies of this type.

Typical calculations showing the theoretical gains due to the inclusion of the regenerative cycle are shown in Figures 6, 7, and 8, and the theoretical gain due to an increase in initial pressure and temperature in Figure 9.

TABLE I: STEAM AND FUEL CONSUMPTION

Operating Conditions					
Shaft horsepower	Port	4250	6367	8500	9350
Propeller rpm	0	67.4	77.75	85	87.7
No. of aux. gen. in operation	2	2	2	2	2
Steam Conditions					
Pressure at turbine throttle, lb gage	440	440	440	440	440
Temperature at turbine throttle, deg F	740	740	740	740	740
Vacuum, in. Hg ref. 30 in. bar.	28.5	28.5	28.5	28.5	28.5
Steam Consumption, Lb/Hr					
Main turbines, without extraction	—	32938	46160	59245	65170
Admit in lieu of extraction	—	1870	2755	3680	4120
Auxiliary turbine generators	5931	4020	4020	4020	4020
Steam auxiliaries	100	100	100	100	100
Air ejectors, main and aux.	250	575	575	575	575
Heating steam	895	970	1020	1070	1120
Evaporators	200	200	200	200	200
Losses (sup. steam)	100	100	100	100	100
Boiler blow to evap.	100	100	100	100	100
Total Lb/Hr	7576	40873	55030	69090	75505
Lb sup. steam per hr	6031	38928	53035	67045	73410
Lb de-sup. steam per hr	1545	1945	1995	2045	2095
Evaporation Per Lb Oil					
Basis: 18,500 Btu oil, boiler eff. 88 per cent					
Superheated steam	13.814	13.814	13.814	13.814	13.814
De-superheated steam	15.83	15.83	15.83	15.83	15.83
Fuel Consumption, Lb/Hr					
Lb oil to evap. sup. steam	436	2810	3870	4850	5310
Lb oil to evap de-sup. steam	98	123	124	130	132
Total Lb/Hr	534	2933	3994	4980	5442
Lb/shp-hr all purposes	—	0.69	0.627	0.586	0.582

TABLE II: ESTIMATED AVERAGE ELECTRIC AUXILIARY LOAD

Apparatus	Hp Each	Tot. No.	No. in Use	CONNECTED LOAD		PEAK LOAD At Sea		PEAK LOAD In Port		24-HR LOAD At Sea	
				Hp	Kw	Factor	Kw	Factor	Kw	Factor	Kw
Main circulating	100	1	1	100	80	0.90	72.5			0.72	58.0
Main condensate	20	2	1	40	35	0.50	8.8			0.45	7.9
Auxiliary circ.	15	3	2	45	39.6			0.60	24.0		
Auxiliary cond.	5	3	2	15	13.2			0.50	4.5		
F.d. blower	50	2	2	100	88	0.40	35.0	.15	13.2	.30	26.4
Main feed	35	2	2	70	61.5	.75	46.0	.30	20.0	.65	40.0
Fire & gen'l serv.	50	1	1	50	44.0	.15	6.6		6.6	.15	6.6
Fire—standby	50	1		50	44.0	0			0	0	
Bilge	15	1	Int.	15	13.2		1.5		.5		.5
Clean ballast	15	1	Int.	15	13.2		1.5		.5		.5
Lub. oil serv.	15	1		15	13.2	.50	6.6			.50	6.6
Fuel oil serv.	7.5	1		7.5	6.6	.70	4.6	.60	4.0	.60	4.0
Air compressor	15	1		15	13.2	.70	9.2	.65	8.6	.65	8.6
Refrigeration	15	1		15	13.2	.30	4.0	.25	3.3	.25	3.3
Lub. oil purifier	2	2		4	3.5		.5		.5		.5
Fuel oil transf.	15	1	Int.	15	13.2		.5		.3		.5
Sanitary	5	1		5	4.4	.4	1.8	4	1.8	.30	1.3
F. W. pump	3	1		3	2.7		1.0		.5		.5
Ref. circ.	2	1		2	1.8		1.5		1.4		1.4
Turning gear	7.5	1		7.5	6.6						
Machine shop	10			10	10.0		1.0		1.0		1.0
Steering gear	30	2		60	53.0		7.5				5.0
Ventilating fans	Misc.			100	88.0		60.0		45.0		45.0
Lighting					30.0		20.0		20.0		20.0
Galley appl.					10.0		3.5		3.1		3.1
Priming pump	1.5	1			1.4		.2		.2		.2
Radio		1			4.5		1.0				1.0
Rotary converter		1			2.0		1.0		1.0		1.0
Battery chg.		1			3.5		.5		.5		.5
Hot water circ.	1	1			.9		.7		.7		.7
Dirty ballast	15	1			13.2		1.0				1.0
Windlass	60	1		60	48.0						
Capstan	35	1			30.0						
Cargo winches	50	16	16	800	700.	30		30	210		
Total Kw					1504.6		298.0		370.7		245.1
Steam consumption lb/hr, 2 units in operation									5931		4020

TABLE III

ESTIMATED AVERAGE STEAM AUXILIARY AND HEATING STEAM LOAD

Apparatus or Use	Port	4250 Shp	6367 Shp	8500 Shp	9350 Shp
Steam Auxiliaries (Intermittent and Standby Service)					
Evaporator feed					
Fire and general service					
Dirty ballast					
Diesel f.o. transfer					
Lub. oil standby					
Sanitary pump					
Aux. feed pump					
Estimated average for group	100	100	100	100	100
Air Ejectors					
Main condensers		325	325	325	325
Auxiliary condensers	250	250	250	250	250
Total for group	250	575	575	575	575
Heating Steam:					
Ship heating, galley, h.w.	695	695	695	695	695
F.o. heaters and tank heating coils	200	275	325	375	425
Total for group	895	970	1020	1070	1120
Evaporator	200	200	200	200	200
Loss—Superheated-Steam	100	100	100	100	100
Boiler Blow	100	100	100	100	100

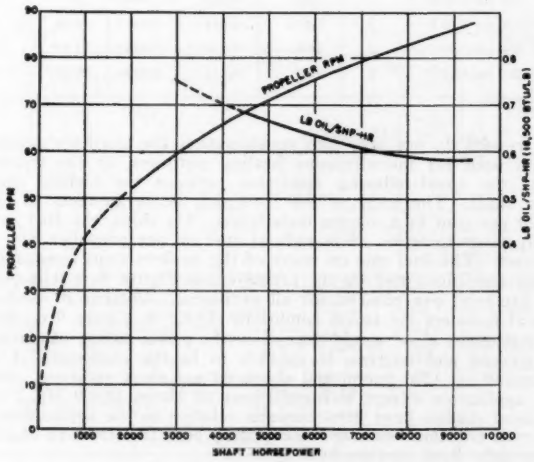


Fig. 5. Calculated fuel-rate curve and propeller speed curve for a 8500-shp turbine-gear-propelled cargo ship

The design factors chosen for the studies shown in Figures 6, 7, and 8 are as follows:

Constant factors:

(a). Heat value of fuel..... 18,500 Btu/pound
(b). Boiler efficiency..... 87 per cent
(c). Steam pressure..... 450 pound g.
(d). Steam temperature..... 740 F.
(e). Vacuum, ref. to 30 inch barometer..... 28.5

Variable factor—Feed heating cycle

(f). Figure 6. Without feed-water heating
(g). Figure 7. One stage of feed heating; final temperature of feed water 230 F.
(h). Figure 8. Three stages of feed-water heating; final temperature feed water 300 F.

The design factors chosen for Figure 9 are the same as for Figure 8, except that the initial steam conditions have been changed as follows:
(i). Figure 9. Initial steam conditions: pressure 1200 pound gauge 950 F.

The final results shown in the calculations are given in Table IV.

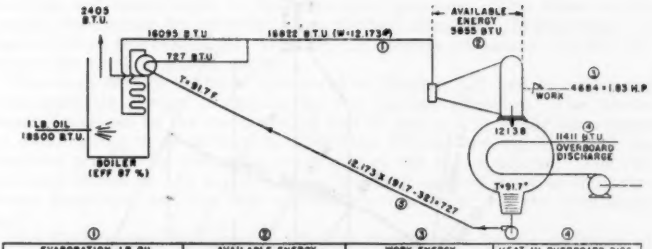
TABLE IV

Study	Steam Conditions	No. of Feed Htr	Hp per Lb Oil, Turbine Shaft	Lb Oil Shp-hr, Turbine Shaft	Btu per Hp-hr	Reduction in Fuel Rate	Heat Rejected to Sea Water Btu per Hp
Fig. 6	450 psi-740 F	0	1.83	0.545	10082	0	6219
Fig. 7	450 psi-740 F	1	1.945	0.514	9509	5.69	5678
Fig. 8	450 psi-740 F	3	2.07	0.483	8935	11.46	5315
Fig. 9	1200 psi-950 F	3	2.30	0.434	8029	20.37	4451

These results do not take into consideration the auxiliary power load, the steam used for miscellaneous heating purposes, or the transmission losses of the speed-reducing apparatus between the turbine shaft and propeller shaft. The total of the foregoing items amounts to approximately 20 per cent in a marine installation. On ships, the fuel rates are invariably given in terms of pounds of fuel oil per propeller Shp.-hr for all purposes. The fuel rate on many of the modern ships operating under the steam conditions and layout presented in Figure 8 is 0.58 pound of (18,500 Btu.) oil per Shp.-hr for all purposes. Whether it would prove economical to select the steam conditions shown in Figure 9 on the basis of fuel economies alone would depend on the power rating of the installation, first cost, and weight. It appears to be the consensus of opinion that pressures of 1200 pound and above do not show sufficient savings to warrant application except in installations of above 10,000 Hp.

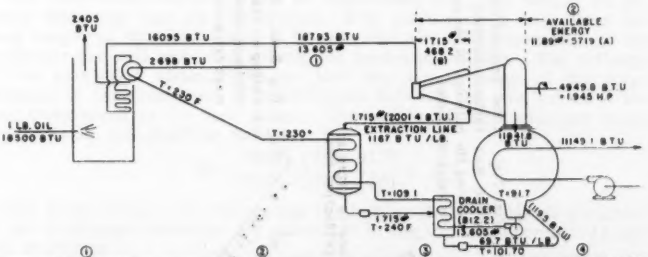
Additional studies have been made in relation to the application of the re-heat cycle and these include re-heating by both the steam re-heat system and separately fired re-superheat system.

Referring to Figure 6, the pounds of steam generated per pound of fuel oil is of simple solution when the heat value of the fuel oil, boiler efficiency, enthalpy of the steam, and enthalpy of the feed water is known.



① EVAPORATION - LB. OIL		② AVAILABLE ENERGY		③ WORK ENERGY		④ HEAT IN OVERBOARD DISC.	
HEAT VALUE FUEL	18500 BTU	REFERENCE - TOTAL HEAT ENTROPY DIAGRAM		TURBINE EFFICIENCY ASSUMED	80 %	TOTAL	16822 BTU
BOILER EFF	87 %	PRESS.	465# ABS	AVAILABLE EN.	5855	WORK 4684	
HEAT AV.	16095 BTU.	TEMP.	740 F	EFF	80	RET'D 727	5411 BTU
TOTAL HEAT 1 LB. ST.	1381 BTU.	VACUUM	28.5 F	WORK EN.	4684 BTU.	TO SEA	11411 BTU.
HEAT IN FEED 91.7 - 32 =	59.7 BTU.	H_1 =	1381 BTU.	HORSE-POWER = 4684 / 2545 =	1.83	⑤ HEAT RET'D IN FEED W	
HEAT TO BE SUPPLIED BY FUEL - 1381 - 59.7 =	1321.3 BTU.	H_2 =	900 BTU	FUEL RATE - LB SHP-HR = 1 / 1.83 =	.545	HOT-WELL T =	91.7
EVAPORATION PER LB. OIL = 16095 / 1321.3 =	12.173 LB.	$H_1 - H_2$ =	481 BTU. LB.			BTU - LB =	32 / 59.7
		TOTAL AVAILABLE EN. = 12.173 X 481 =	5855 B.T.U.			LBS	12.173
						HEAT RET'D	727 B.T.U.

Fig. 6. Heat distribution in a marine power plant without feed-water heating. Steam conditions: 450 lb gage, 740 F, 1½ in. abs back pressure



① EVAPORATION - LB. OIL		② AVAILABLE ENERGY		③ WORK ENERGY		④ HEAT DV. B'D DISCHARGE	
HEAT VALUE FUEL	18500 BTU	REFERENCE - TOTAL HEAT - ENTROPY DIAGRAM		TURBINE EFF	80 %	TOTAL =	18793.0
BOILER EFFIC	87 %	(A) 450# - 740 F -	28½ VAC	(A) AV. EN. =	5719	RET'D DRIPS	119.5
HEAT AVAILABLE	16095 BTU	H_1 =	1381 BTU.	(B) AV. EN. =	468.2	TOTAL	18912.5
TOTAL HEAT 1 LB ST.	1381 BTU	H_2 =	900 BTU	TOTAL	6187.2	WORK	4949.8
HEAT IN F.W. 230 - 32 =	198	$H_1 - H_2$ =	481 BTU	TURBINE EFF	80	EXTR	2001.4
HEAT TO BE SUPPLIED BY FUEL	1183	TOTAL - 11.89 X 481 =	5719	WORK ENERGY	4949.8 BTU	COND.	812.2
EVAPORATION PER LB. OIL = 16095 / 1183 =	13.605 LB.	(B) 450# - 740 F -	25# ABS	HORSE POWER = 4949.8 / 2545 =	1.945	TO SEA	7763.4 / 11149.1 BTU
		H_1 =	1381 BTU				
		H_2 =	1108 BTU	FUEL RATE = 1 / 1.945 =	.514		
		$H_1 - H_2$ =	273 BTU				
		TOTAL = 1.715 X 273 =	468.2				

Fig. 7. Heat distribution in power plant with single-stage extraction for feed-water heating. Steam conditions: 450 lb gage, 740 F, 1½ in. abs back pressure

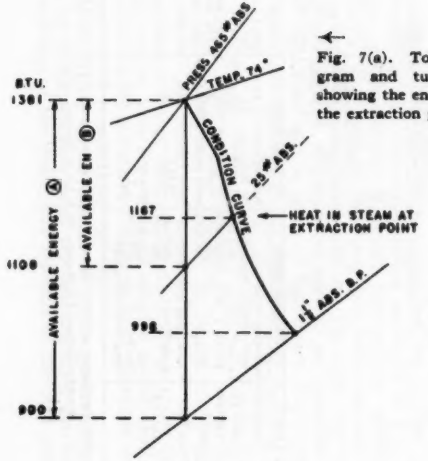


Fig. 7(a). Total heat-entropy diagram and turbine-condition curve showing the enthalpy of the steam at the extraction point

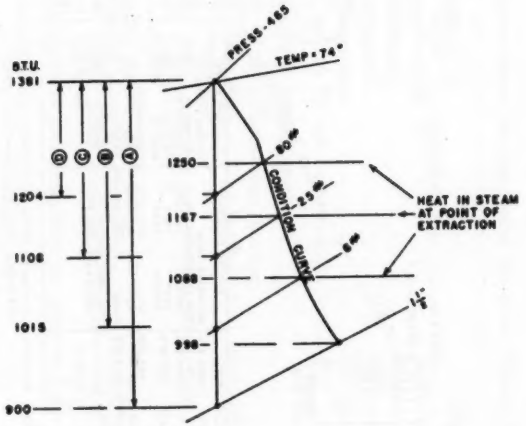


Fig. 8(a). Total heat-entropy diagram and condition curve for three-stage extraction

The calculation in Section (2) of Figure 6 shows the available energy in one pound of steam when expanded adiabatically from the given initial pressure and temperature to the given back pressure. This value can be quickly determined by referring to a Mollier Diagram. Multiplying this figure which represents (H_1-H_2) by the pounds of steam (12.173 lb.) equals the total available energy.

The work energy is shown calculated in Section (3) and is simply the total available energy multiplied by the turbine efficiency. The horsepower developed by the one pound of fuel oil can now be easily determined by dividing the work energy by 2545 (the Btu. in one Hp-hr.), and the fuel rate per Hp. by taking the reciprocal of the Hp. produced. The calculations shown in (4) and (5) show the heat carried away in the overboard discharge, and the heat returned to the boiler via the feed-water path.

The study shown in Figure 7 although complicated by the inclusion of one feed-water heater using extraction steam from the main turbine is not difficult of solution. Calculation (1) is made in a similar manner to (1) in Figure 6. The higher feed-water temperature naturally increases the pounds of steam evaporated per pound of fuel oil. Assuming that the feed-water heater is of the closed type and designed on the basis of a heat head of 10 F. to effect a heat transfer it will be necessary to maintain a pressure of 25 pound abs (240 F.) to attain a feed-water temperature of 230 F. A drain cooler will also be required to absorb at least a part of the heat contained in the drips before returning such drips to the condenser hot-well. It has been assumed that the drain cooler has also been designed on the basis of a 10 F. heat head.

A turbine-condition curve may now be drawn showing the available energy in the steam for both pressure drops, that is, from the initial pressure and temperature to 1½ inch abs back pressure, and from the initial pressure and temperature to the extraction pressure of 25 pound abs. Such a diagram, showing the enthalpy of the steam at the extraction point, is shown in Figure 7 (a).

The total work energy cannot be determined until the weight of the steam following each path is known. This calculation is easy since the total weight of the feed water is known, and also its temperature at the condenser hot-well and when leaving the feed-water heater. The enthalpy of the extraction steam is known and also the enthalpy of the drips returned to the condenser. The difference between the two values is the heat transferred to the feed water. The weight of the extraction steam is now easily calculated as follows:

$$W = \frac{13.605\ (230-91.7)}{1167-(101.7-32)} = 1.75 \text{ lb.}$$

The work energy (hp produced at the turbine shaft) and heat discarded in the overboard discharge are calculated in the same manner as (3) and (4) in Figure 7.

The study shown in Figure 8 is more in line with modern practice on ships. It will be noted that a direct-contact feed-water heater is used to boost the temperature of the feed water to 240 F. before it enters the high-pressure heater. This heater is of the de-aerating type.

A heat balance of this type resolves itself into a circular or indeterminate problem.

When a heat balance is computed for a complete power plant operating at a specific power rating it is necessary to first estimate the total weight of the feed water before the weight of the extraction steam can be determined. In the system shown in Figure 8 this total quantity is a fixed value dependent only upon the initial steam conditions, feed-water temperature, and boiler efficiency. This places the power produced on a variable basis.

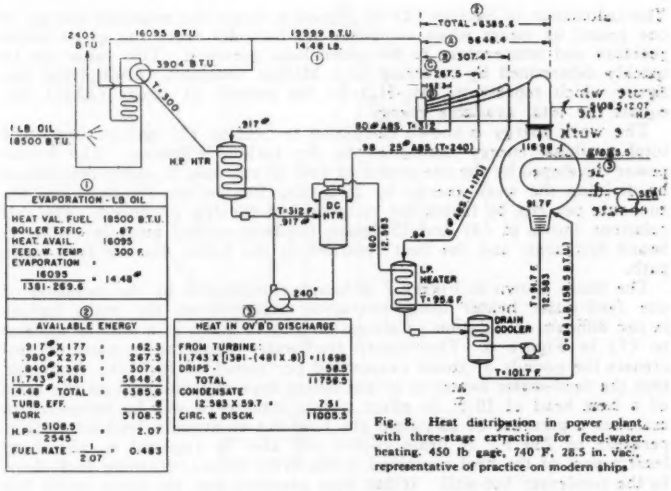


Fig. 8. Heat distribution in power plant with three-stage extraction for feed-water heating. 450 lb gage, 740 F, 28.5 in. vac., representative of practice on modern ships

In determining the weight of the extraction steam the quantity required by the high-pressure heater is subject to direct calculation as both the weight of the feed water and the temperature in and out of the heater is known.

The intermediate or direct-contact heater is not subject to direct calculation as the weight of the intermediate extraction steam affects the

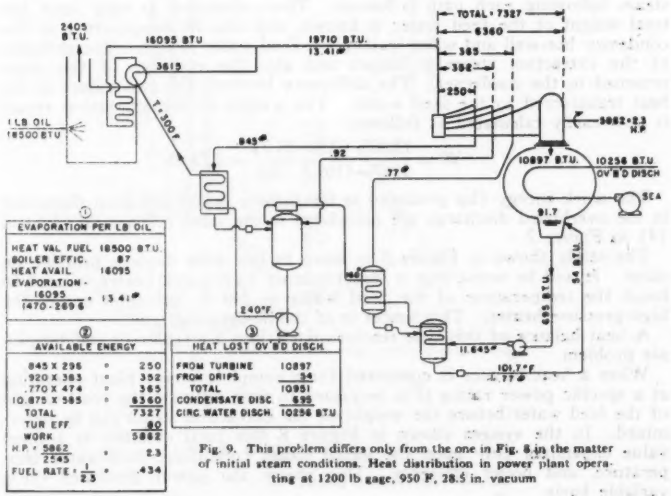


Fig. 9. This problem differs only from the one in Fig. 8 in the matter of initial steam conditions. Heat distribution in power plant operating at 1200 lb gage, 950 F, 28.5 in. vacuum

weight of the feed-water entering from the low-pressure heater. The enthalpy of the extraction steam is determined from the turbine-condition curve as shown in Figure 8 (a) and the enthalpy of the liquid from the low-pressure heater from a book of steam tables. As a general rule, the weight values entering from both sources may be quickly estimated after two trials.

Having established the weight of flow from the intermediate extraction point the weight of flow passing through the low-pressure heater and drain cooler now becomes a fixed value. The weight of the low-pressure extraction steam therefore becomes subject to direct calculation. Having established the weight of flow through each path of the turbine, the power produced is subject to calculation the same as in the other examples shown.

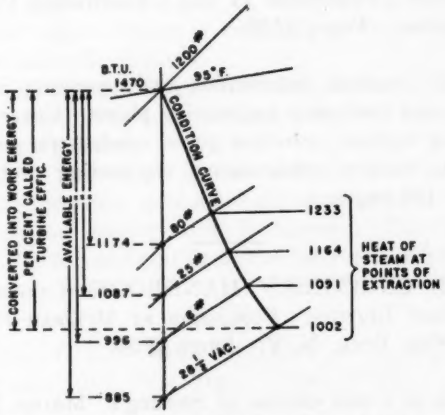


Fig. 9(a). Total heat-entropy diagram and condition curve for conditions shown in Fig. 9

The problem shown in Figure 9 differs from Figure 8 only in the matter of the initial steam conditions. It is calculated by the same methods previously shown. The calculated fuel rates per Shp.-hr for all purposes for ships operating under the conditions shown in Figure 9 range from 0.48 to 0.52 pound depending upon the extent of the auxiliary power needs.

Future of Steam Power Plants.

Steamships of the type under consideration burn a very low grade of fuel oil known as bunker C. This type of fuel oil, although high in heat content, is both cheap and plentiful. In making comparisons with other forms of drive which have a lower fuel rate but which require a refined fuel oil the cost differential may still favor the steam plant. This can be determined only when the trade route of the ships is known and the relative costs of the fuel oil at ports of call.

As to the future trends in marine engineering practice it is unsafe to predict too far into the future especially when one views the progress made in the past. Continuous research and development is the only way in which one can be certain to remain on top in this most interesting field. Romance of the sea and romance of engineering have joined hands.

BOOK REVIEWS.

PRACTICAL BOOK FOR MACHINISTS AND ENGINEERS, BY ROBERT F. BYRNES, LICENSED MARINE AND STATIONARY ENGINEER. PUBLISHED BY THE CHRISTOPHER PUBLISHING HOUSE, BOSTON. PRICE $2.00.

Condensed practical information and operating instructions for marine and stationary engineering plants. Covers turbines, reciprocating engines, reduction gears, condensers, pumps, electrical plants, boilers, refrigeration, engineering problems, and fire rules. 156 pages.

MARINE ENGINEERS' HANDBOOK—LABBERTON AND MARKS—FIRST EDITION. PUBLISHED BY MCGRAW-HILL BOOK CO., INC., NEW YORK, N. Y. PRICE $7.50.

This book is a new edition of Sterling's "Marine Engineers' Handbook", brought up to date under the editorship of Professor J. M. Labberton. It contains much valuable material on engineering fundamentals reprinted from the Fourth Edition of the "Mechanical Engineers' Handbook" with the cooperation of Professor Lionel S. Marks. With the exception of the section on reciprocating steam engines all sections on special engineering topics are completely new.

It contains a wealth of information on engineering fundamentals, mathematics, mechanics, thermodynamics, hydraulics, boilers, all types of engines, electrical engineering, auxiliary machinery, refrigeration, ships' trials, etc., written by a staff of specialists in various fields.

Design engineers, technical students, operating engineers and others will find this book a useful tool and a guide to the many and rapid developments of recent years in this field.

TABLE OF ARC SIN X, TABLES OF ASSOCIATED LEGENDRE FUNCTIONS. PREPARED BY THE MATHEMATICAL TABLES PROJECT CONDUCTED UNDER THE SPONSORSHIP OF THE NATIONAL BUREAU OF STANDARDS. PUBLISHED BY COLUMBIA UNIVERSITY PRESS, NEW YORK. PRICE: TABLE OF ARC SIN X, $3.50. TABLE OF ASSOCIATED LEGENDRE FUNCTIONS—$5.00.

Whoever has tried to apply mathematics to engineering knows the pressing need for a store of mathematical tables. The Mathematical Tables Project is rendering an important service in the task of scientific research.

The Table of Arc Sin X gives 12-place values of the arc sin in radian measure. The book contains tables and formulas used in accurate interpolation.

The Table of Legendre functions will fill an important gap in the existing literature, since more and more problems involving these functions are coming to the foreground. This table gives the functions to 6 significant figures at intervals of 0.1.

UNITED NATIONS GOVERNMENT, BY AMOS J. PEASLEE. G. P. PUTNAM'S SONS, NEW YORK. PRICE $2.00.

This book is a sequel to an earlier work "A Permanent United Nations charter, and a summary of the present status of inter- the San Francisco conference, a critical analysis of the United Nations charter, and a summary of the present status of international organization. In it the author points out the defects in the League of Nations and expresses his opinion that the present United Nations organization is merely a temporary phase of world government, similar to our Articles of Confederation, and that it will be followed by a more effective international organization similar in general to our constitutional government.

CARGO DEADWEIGHT DISTRIBUTION, BY MARTIN HEUVELMANS. PUBLISHED BY CORNELL MARITIME PRESS, NEW YORK, N. Y. · PRICE $1.50.

A short practical book on cargo stowage intended for cargo planners, ship's officers, and stevedors. Explains stowage of cargo to achieve good stability and trim, calculation of G.M., space-saving methods, loading plans, etc., applied to typical situations.

THE GREAT PACIFIC VICTORY, BY GILBERT CANT. PUBLISHED BY JOHN DAY COMPANY, NEW YORK, N. Y. PRICE $3.50.

This book is a sequel to "The War at Sea" and "America's Navy in World War II" by the same author. In this volume the author covers thoroughly the operations on sea, land, and air from the Solomons campaign to the final surrender. Beginning with a "snapshot" of the strategic scene in February 1943, this book goes on to encompass our converging sweep towards Japan: the occupation of the Russells, the Solomons offensive and mop-up, the carrier strikes at islands ever nearer to the goal, the closing of the pincers on Rabaul, Tarawa, the Marianas, the Philippines, Iwo and Okinawa, the bombing of Japan and the final surrender.

Four chapters are devoted to profiles of Admirals Nimitz, Halsey, Spruance, and Mitscher. The author, formerly war editor of the New York Post and a war editor of Time magazine, is well qualified to write such a book.

ASSOCIATION NOTES.

Addresses.

Correct addresses are becoming more and more important. It is urged that members keep the Society informed. This is the only way you can be certain of receiving your Journal and other mail.

Election of Officers.

The following have been elected officers of the Society for the calendar year 1946:

President:

Rear Admiral Harvey F. Johnson, U.S.C.G.

Secretary-Treasurer:

Captain F. W. Walton, U.S.N.

Council:

Rear Admiral S. S. Kennedy, U.S.N.
Rear Admiral R. W. Paine, U.S.N.
Captain Logan McKee, U.S.N.
Captain W. D. Chandler, U.S.N.
Lieut. Commander K. H. Beeson, U.S.N.R.
Commander R. T. Simpson, U.S.N.R.
Captain G. A. Taylor, U.S.C.G.
Mr. R. E. Post
Mr. F. C. MacKrell

Amendments to By-Laws.

The amendments to the By-Laws recently proposed have been adopted by a vote of 931 to 11.

Annual Banquet.

The Council of Society is happy to announce the resumption of the Annual Banquet, after a lapse of five years due to war conditions, on Friday, April 5, 1946 at the Statler Hotel, 16th and K Streets, N.W., Washington, D. C.

This occasion is offered by the Society for those of the service and industry to meet and continue the alliances which have prevailed during the past and have contributed so much to Victory.

It is planned to present speakers who will bring to you the significance of engineering during the War and for the future.

Attendance.

It is regretted that, due to limited accommodations, it is necessary to restrict attendance, giving preference to members. Reservations will be considered in the order of receipt under the following conditions:

a. Requests for reservations for *Member* guests and the desired number of non-member guests must be received by March 1, 1946, listing names of *member* guests only. Check must accompany request for reservations in full.

b. Reservations desired for non-members, not to exceed one for each *Member* in a party, must be received with the request for reservations for *Members,* giving only the total number desired. If member reservations permit, requests for non-member reservations will be accepted or proportionately allocated. Applicants will be notified immediately after March 1, 1946 of the number of non-member reservations allocated. The name of each non-member guest must be received by the Society by March 20, 1946. Refund will be made for any over payment as soon as practicable.

Reservation forms are enclosed.

Tables will seat 12 each.

Seating arrangements may be specified for parties. The Society reserves the right to make changes in seating arrangements without notice.

The price is $7.50 per plate for *members* and $10.00 per plate for *non-members.* Make check payable to "American Society of Naval Engineers."

Cancellations must be received by March 29, 1946.

INFORMATION.

Please wear uniform or dinner coat.

Lists of members will be furnished upon request.

Guests desiring accommodations at the hotel should address the Manager, Statler Hotel, Washington, D. C., mentioning

attendance of the Banquet. Reservations may be expected if requests are received prior to March 15.

Guests are invited to the Continental Room on the second floor 6:30 to 7:30 where friends will be found and cocktails served.

Seating lists will be available. Seating will be by table number and place card.

It is requested that guests proceed to the Banquet Room at 7:30 so that dinner may be served at 8:00.

All gratuities at tables will be assumed by the Society.

MEMBERSHIP.

Attention of all non-member officers is especially invited at this time to the advantage and benefit to be attained by membership in this Society. The Society was organized in 1888—for the sole purpose of the advancement of Engineering. Its JOURNAL, published quarterly, In February, May, August and November, and completing its 57th year of continuous publication, has attained a foremost rank in its field and is a recognized authority throughout the engineering world. It is used generally by Naval officers and civilians alike as a reliable reference book. The Bureau of Ships considers the JOURNAL of such value that it makes it available for the libraries of all Naval vessels and shore activities of any size. The presence of the JOURNAL in your personal library can not fail to be of great benefit to you, often making available information of great value which can be obtained from no other source.

It is particularly suggested that officers of the Naval Reserve, who may be contemplating return to civil life, should arrange for membership prior to detachment from active duty.

Annual dues are $5.00, which are probably lower than those of any other technical Society. There is no initiation fee, nor any extra charge for the JOURNAL. This extremely low cost is possible only because the Society is bona fide operated solely for your benefit without any pecuniary profit to any one.

An application blank generally will be found at the back of each copy of the JOURNAL. However, if none is found, application by letter will be accepted.

It is suggested that Members will confer a real favor on non-members by bringing the Society to their attention.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine, dignified insignia. It is one-half inch in diameter. The oak leaves and lettering are red on a gold back-ground. It is available to all members at fifty cents (50c) each.

Membership Additions.

The following have joined the Society since the publication of the November, 1945, Journal. Addresses of personnel on ships not given.

Naval.

Alderman, O. B., Jr., Lieut., j.g., U. S. N. R.,
2701 First Ave., S., St. Petersburg, Fla.

Anderson, Earle C., Ensign, U. S. N. R.,
Engineering Student, Lehigh University
Mail 6829 Narrow Ave., Brooklyn, N. Y.

Bailey, G. W., Captain, U. S. N.,
care Supervisor of Shipbuilding, U. S. N., Quincy 69, Mass.

Ball, Warren Wilson, Lieut., j.g., U. S. N. R.

Barbiere, Henry V., Commander, U. S. C. G. R.,
Officer in charge, Marine Inspection, 12th Naval District
Mail 27 Atalya Terrace, San Francisco 17, Calif.

Bean, John R., Lieut., j.g., U. S. N. R.

Benson, R. H., Commander, U. S. N.,
2016 E. Brae Burn Road, Altadena, Calif.

Birkhimer, Frederick E., Ensign, U. S. N.,
Country Club, Salem, Ohio
Boaz, Richard M., Commander, U. S. N., Ret.
care Fairbanks, Morse & Co., 1427 Eye St., N.W.,
Washington, D. C.
Bornwasser, Louis Philip, Lieut., j.g., U. S. N. R.,
2040 Douglas Boulevard, Louisville, Ky.
Bower, Edward T., Lieut. Commander, U. S. N.,
215 King George St., Annapolis, Md.
Boyle, R. E., Lieut., j.g., U. S. N. R.
Bradley, Vincent Arthur, Chf. Mach., U. S. N.
Broyderick, N. L., Lieut. Commander, U. S. N. R.,
61 Neptune St., Lynn, Mass.
Brunet, Robert D., Lieut., U. S. N. R.,
Atlantic, Gulf, Pacific Co., 15 Park Row, New York, N. Y.
Burke, John F., Jr., Lieut., U. S. N. R.,
5809 Sixth Ave., Brooklyn, N. Y.
Caldwell, James Howard, Lieut., j.g., U. S. N.,
2218 Albatross St., San Diego 1, Calif.
Carlberg, Edward F., Lieut., j.g., U. S. N. R.
Chalmers, Norman E., Lieut. Commander, U. S. N. R.
Collender, Edward H., Lieut., U. S. N. R.,
3055 Leland Ave., Chicago, Ill.
Collins, Francis T., Lieut., U. S. N. R.,
211 Crooke St., Green Bay, Wis.
Cooke, George E., Lieut., j.g., U. S. N.
Craig, Donovan D., Elect., U. S. N. R.
Crial, Richard S., Lieut., U. S. N. R.,
P. O. Box 114, Des Moines, Wash.
Cusack, John J., Lieut., j.g., U. S. N. R.,
26 Corona St., Boston, Mass.
Dadetta, Arthur P., Lieut., j.g., U. S. N. R.,
199 Adams St., Rochester, N. Y.
Dalton, Carl M., Captain, U. S. N.,
Code 445, Bureau of Ships, Navy Dept., Washington, D. C.
Dalton, William Robinson, Lieut., U. S. N. R.
Davies, Ivor T., Lieut., U. S. N. R.,
723 N. E. 1st Ave., Portland, Oregon

Davis, Lloyd A., Commander, U. S. N. R.,
P. O. Box 3006, Savannah, Ga.
Davis, William V., Chf. Mach., U. S. N. R.,
86 So. West 9th St., Ontario, Oregon.
Dign, George N., Lieut., U. S. N. R.,
99 West Commerce St., Bridgeton, N. J.
Donohue, T. F., Captain, U. S. N.
Droege, Raymond B., Chf. Mach., U. S. N.
DuVall, Richard, Ensign, U. S. N. R.,
4108 Coolidge Ave., Oakland 2, Calif.
Elliott, G. T., Lieut. Commander, U. S. N.
Ewing, Edgar A., Commander, U. S. N. R.,
2007 O St., N.W., Washington, D. C.
Flint, Morris B., Lieut. Commander, U. S. C. G.,
1405 K St., S.E., Washington 3, D. C.
Gaidis, Peter Northrop, Jr., Lieut., U. S. N. R.,
55 Green St., Jamaica Plain, Mass.
Gawlockl, Lester R., Lieut., U. S. N. R.,
40 W. Neptune St., Lynn, Mass.
Gibbs, Braman, Lieut., U. S. N.,
18 Park St., Hudson, Mass.
Goodman, Robert Barber, Jr., Lieut. Commander, U. S. N. R.
Gray, Theodore F., Lieut. Commander, U. S. N. R.,
671 State St., Portsmouth, N. H.
Griep, Elmer F., Commander, U. S. N. R.,
752 Grisley Peak Boulevard, Berkeley, Calif.
Haaren, Craig F., Lieut., U. S. N. R.,
230 Corlies Ave., Allenhurst, N. J.
Haase, R. W., Sales Engr.,
care Foster Wheeler Corp., 165 Broadway, New York, N. Y.
Haase, Wynne W., Lieut. Commander, U. S. N. R.,
Box 211, Martinsville, Ind.
Hague, W. M., Captain, U. S. N.,
Room T4-2001, Bureau of Ships, Navy Dept.,
Washington, D. C.
Hamilton, H. L., Lieut. Commander, U. S. N. R.,
Room 702, 917 15th St., N.W., Washington 3, D. C.
Hartford, Arnold A., Lieut. Commander, U. S. N. R.,
25 Cedar St., Haverhill, Mass.

Hayes, John A., Captain, U. S. N., Ret.,
Code 651, Bureau of Ships, Navy Dept., Washington, D. C.
Healey, William James, Lieut., j.g., U. S. N. R.,
324 Tinosa Ave., Vallejo, Calif.
Hemley, E. A., Lieut. Commander, U. S. N.,
Postgraduate School, Naval Academy, Annapolis, Md.
Herman, Neil B., Lieut., U. S. N. R.
Hess, Walter John, Jr., Ensign, U. S. N. R.
Hewitt, John Vance, Jr., Lieut. Commander, U. S. N. R.,
970 Park Ave., New York, N. Y.
Hochner, Royal Monroe, Lieut., j.g., U. S. N.
Hoffman, James L., Lieut., j.g., U. S. C. G. R.,
The Gables, Larchmont, N. Y.
Hogg, J. V., Lieut. Commander, U. S. N. R.,
32 Grant Ave., Amsterday, N. Y.
Horner, Wesley L., Lieut., U. S. N.
Hreczuck, Walter M., Lieut., j.g., U. S. C. G. R.,
454 3d St., Brooklyn, N. Y.
Jackson, Albert J., Lieut., U. S. N. R.,
1425 So. Broad St., Trenton, N. J.
Jamison, John J., Jr., Lieut., U. S. N. R.,
909 East Pearl St., Butler, Pa.
Jennings, Thomas G., Ensign, U. S. N. R.,
214 St. Johns Place, Brooklyn 17, N. Y.
Jewett, Robert V., Ensign, U. S. N. R.,
RR No. 1, Powell, Ohio
Kazor, Walter R., Ensign, U. S. N. R.,
Kellogg, Robert B., Jr., Lieut., j.g., U. S. N. R.,
78 Cortland St., Norwich, N. Y.
Kingsler, George D., Jr., Lieut., j.g., U. S. N. R.,
Hemphill Diesel School, 3-28 Queens Boulevard,
Long Island City, N. Y.
Mail 451 40th St., Brooklyn, N. Y.
Kniskern, L. A., Captain, U. S. N.,
Bureau of Ships, Navy Dept., Washington, D. C.
Koerner, Frank A., Chf. Carp., U. S. N.,
228 East Timbar St., Pontiac, Ill.
Kramer, Bernard L., Lieut., j.g., U. S. N. R.,
22 Ramsey Place, Albany, N. Y.

Krenitsky, Andrew, Lieut., U. S. N. R.,
11 Janette Ave., Binghamton, N. Y.
Lair, Rodney B., Captain, U. S. N.,
1651 N. Greenbrier St., Arlington, Va.
LaRose, Edgar H., Ensign, U. S. N.,
Route No. 2, Oakley, Idaho
Larson, Carl O., Lieut., j.g., U. S. C. G. R.,
624 Chestnut St., Lynn, Mass.
LaSpada, Jack Anthony, Lieut. Commander, U. S. N.,
care Senior Naval Officer, Mass. Inst. of Tech.,
Cambridge 39, Mass.
Levine, A. B., Lieut., U. S. N. R.,
31 Ocean Parkway, Apt. 1 L, Brooklyn 18, N. Y.
Levitt, Eugene, Ensign, U. S. N. R.
Lindsey, Ray W., Lieut., j.g., U. S. N. R.
Lowell, James M., Lieut., U. S. C. G. R.,
505 Quackenbosh St., N.W., Washington, D. C.
MacKay, Hugh, Captain, U. S. N.,
903 Beverly Drive, Alexandria, Va.
McClure, George W., Lieut., U. S. N. R.,
Office of Chief of Naval Operations, Navy Dept.,
Washington, D. C.
Main, Archibald M., Jr., Ensign, U. S. N. R.,
Bath Iron Works Corp., Bath, Maine
Mail 3519 Valley Drive, Alexandria, Va.
Marika, Andrew, Chf. Carp., U. S. N. R.
Mattes, John C., Ensign, U. S. N. R.
May, George E., Lieut., j.g., U. S. N. R.
Murati, George T., Lieut. Commander, U. S. C. G.
Murphy, Moyes J., Lieut. Commander, U. S. N. R.,
1554 Mt. Eagle Place, Alexandria, Va.
Murphy, William D., Lieut., U. S. N. R.,
1020 7th St., So., Fargo, N. Dakota
Nelson, Harlan, Lieut., U. S. N. R.
Newell, John R., Lieut., j.g., U. S. N. R.,
1080 Washington St., Bath, Maine
Nordensen, Bengt K., Lieut., j.g., U. S. N. R.,
American Locomotive Co., Barr Building, Washington, D. C.

O'Donnell, Lawrence H., Lieut., j.g., U. S. N. R.,
532 16th St., Oakland, Calif.
Parks, Raymond D., Lieut., j.g., U. S. N.,
Box 101, South, Plainfield, N. J.
Pospisil, Robert, Ensign, U. S. N. R.,
31-70 43d St., Astoria, L. I., N. Y.
Racine, Arthur J., Ensign, U. S. N. R.
Richards, Frank G., Lieut., U. S. N. R.
Ritch, John B., Jr., Lieut. Commander, U. S. N.,
2877 So. Buchanan, Apt. A2, Fairlington, Va.
Roper, John, Lieut., U. S. N.,
Boiler Design Div., Foster Wheeler Corp.
Mail, 36 E. 64th St., New York 21, N. Y.
Roshong, Raymond G., Engineer, So., Calif. Branch,
Crane Packing Co.,
Mail 4855 Gentry Ave., North Hollywood, Calif.
Rouse, A. M., Jr., Lieut., j.g., U. S. N. R.
Row, Gerald G., Ensign, U. S. N. R.
Sackmann, John Richard, Commander, U. S. N.,
care FPO, NOB, Norfolk, Va.
Sala, Earl V., Jr., Lieut., j.g., U. S. N. R.
Schumacher, T. L., Captain, U. S. N.,
Room 4830, Dept. of Commerce Building, Washington, D. C.
Senser, Lisle H., Jr., Ensign, U. S. N. R.
Seutter, Carl O., Lieut., j.g., U. S. N. R.,
4126 247th St., Little Neck, N. Y.
Shafer, Clifford A., Lieut., j.g., U. S. N. R.
450 N. 34th Boulevard, Camden, N. J.
Shure, William H. N., Lieut., j.g., U. S. N. R.,
6009 Greenspring Ave., Baltimore, Md.
Smith, Harlen E., Lieut., j.g. U. S. N.,
2240 16th St., Parkersburg, W. Va.
Smith, Johnson H., Lieut., U. S. N. R.,
309 Sandy Bank Road, Westover Hills, Media, Pa.
Sosa, Ferdinand, Ensign, U. S. N. R.,
1793 Sedgwick Ave., Bronx, New York, N. Y.
Steuernagel, Carl A., Lieut., j.g., U. S. N. R.,
1227 Forest Ave., Staten Island, N. Y.

Stiansen, Steve, Jr., Chf. Mach., U. S. N.,
25 Mason Ave., Newton, N. J.

Storrs, William H., Lieut., U. S. N. R.,
care The Chapman Valve Mfg. Co., 165 Congress St.,
Boston 10, Mass.

Sweet, P. M., Lieut., j.g., U. S. N. R.

Sylvester, E. W., Captain, U. S. N.
Bureau of Ships, Navy Dept., Washington, D. C.

Tiedeman, Nicholas, Lieut. Commander, U. S. N. R.,
509 Tennessee Ave., Alexandria, Va.

Thornton, William E., Lieut., U. S. N. R.,
Audit Sect., MRDO, Room 300, 131 W. Jackson Boulevard,
Chicago 4, Ill.

Titcomb, Andrew A., Lieut., j.g., U. S. N. R.

Updyke, Robert F., Ensign, U. S. N. R.,
809½ N. Spadra Ave., Fullerton, Calif.

Verseput, Herman Ward, Lieut., U. S. N.,
712 Hawthorne St., NE., Grand Rapids 3, Mich.

Vivian, William W., Lieut., j.g., U. S. N. R.,
20228 Danbury Ave., Detroit, Mich.

Wade, Edwin J., Lieut., U. S. N. R.,
5004 So. Belmont, Minneapolis, Minn.

Wagner, John J., Chf. Elect., U. S. N. R.,
46035 190 St., Flushing, L. I., N. Y.

Walker, William M., Lieut. Commander, U. S. N. R.

Walter, John C., Lieut. Commander, U. S. N. R.,
3130 Wisconsin Ave., N.W., Washington 16, D. C.

Wax, Bernard M., Lieut. Commander, U. S. N. R.,
Inspector of Mach., care Electro Motive Div., G.M.C.,
La Grange, Ill.
Mail 100 Walnut Hill Road, Newton Heights, Mass.

Waxman, Robert Clarke, Ensign, U. S. N. R.

Wills, Oren Marshall, Lieut., U. S. N. R.

Wilson, Lawrence O., Ensign, U. S. N. R.

Wimer, Roy Martin, Lieut., U. S. N. R.,
2009 E. 126th St., Willowbrook, Calif.

Wolfe, Lindsay H., Lieut., U. S. N. R.,
410 Walnut Road, Ben Avon, Pittsburgh, Pa.

Young, J. Virgil, Lieut., j.g., U. S. N. R.

Young, Parke G., Lieut., U. S. N., Ret.,
Allis Chalmers Mfg. Co., 621 Southern Building,
Washington 5, D. C.

Zeigler, George E., Jr., Lieut. Commander, U. S. N. R.,
3914 Cloverhill Road, Baltimore 18, Md.

CIVIL.

Carroll, Stanley O., Senior Engineer, Electrical, Bureau of Ships,
Navy Dept., Washington, D. C.
Mail 210 E. Bellefont Ave., Alexandria, Va.

Condon, Joseph F., Electrical Engineer P5, Bureau of Ships,
Navy Dept., Washington, D. C.
Mail 3301 Stephenson Place, N.W., Washington, D. C.

Evenson, Elwood, Naval Architect, Supervisor of Shipbuilding,
U. S. N., Bethlehem Steel Co., Staten Island, Calif.
Mail 25 Deems Ave., Westerleign, S. I., N. Y.

Falk, L. W., Executive Engineer, The Falk Corporation,
Milwaukee 8, Wis.

Fletcher, Lindsay C., Steam Application Engineer, Westinghouse
Electric Corp., Lester, Pa.

Harmon, Ralph N., Manager, Engrg. Dept. & Industrial
Electronics Div., Westinghouse Electric Corp.,
2519 Wilkens Ave., Baltimore, Md.

Hathaway, Herbert F., Engineer, General Electric Co.
806 15th St., N.W., Washington 5, D. C.

Hazard, Henry N., Asst. to Gen. Engineer, Boston District,
Bethlehem Steel Co.,
Mail, 103 Hobart Ave., Braintree, Mass.

Kessler, George W., Asst. Head, Marine Design & Estimating
Div., The Babcock & Wilcox Co.
Mail, 1594 Metropolitan Ave., The Bronx, New York, N. Y.

King, Hubert G., Diesel Engine Governor Service Manager,
Marquette Metals Products Co., 47 Wooster St.,
New York, N. Y.

Knapp, Harold D., Manager, Sonar Section, Engineering
Products Dept., Radio Corp. of American, RCA Victor Div.
Building 15-4, Camden, N. J.

Lawrence, Calvin H., Engineer Marine Dept., The Babcock & Wilcox Co., 85 Liberty St., New York 6, N. Y.

Leland, Marshall B., Head, Marine Design & Estimating Div., Marine Dept., The Babcock & Wilcox Co., 85 Liberty St., New York 6, N. Y.

Lindauer, Edwin R., Electrical Engineer P-6,
Bureau of Ships, Navy Dept.
Mail, 1136 N. Frederick St., Arlington, Va.

McCrary, Samuel E., Senior Engineer, Bureau of Ships, Navy Dept.
Mail 422 E. Columbia St., Falls Church, Va.

Mahoney, James B., Marine Engineer, Code 445,
Bureau of Ships, Navy Dept., Washington, D. C.

Myers, Walter F., Engineer, Cooper-Bessemer Corp.,
440 Investment Building, Washington, D. C.

Peterson, J. E., Vice President, General Machinery Corp.,
Hamilton, Ohio.

Potter, Edwin E., Commercial Vice President, General Electric Co., 806 15th St., N.W., Washington, D. C.

Reed, Pinckney B., Govt. Sales Section, Radio Corp. of America, 1625 K St., N.W., Washington, D. C.

Ruby, William N., Jr., Washington Sales Representative,
Cleveland Diesel Engine Division, G. M. Corp.,
504 Hill Building, Washington, D. C.

Sanford, Edward R., Engineer of Tests, Penn Steel Castings Co., Chester, Pa.

Speer, James W., Consulting Engineer, Carswell Marine Associates
Lake Shore Engine Co., 15 Park Row, New York, N. Y.
Mail, Box 474, Easton, Md.

Steele, Charles Richard, Installation & Service Engineer,
Cleveland Diesel Engine Division, G. M. Corporation,
2800 E. 122d St., Cleveland 20, Ohio.

Stine, W. E., Senior Engineer, P5, Bureau of Ships, Navy Dept.
Mail, 7218 Cobalt Road, Washington 16, D. C.

Summerfield, Marvin F., Application Engineer, Marine & Industrial, Fairbanks, Morse & Co.
Mail, 1606 N. Rhodes St., Apt. C-2, Arlington, Va.

Tichvinsky, L. M., Dept. Professor of Mechanical Engineering, Washington University, St. Louis 3, Mo.

Walbridge, John B., Naval Machinery Section Manager,
Westinghouse Electric Corporation,
Mail, 1012 Drexel Ave., Drexel Hill, Pa.

ASSOCIATE.

Andrus, Carl F., Salesman, Cargocaire Engineering Corp.,
15 Park Row, New York, N. Y.

Arnson Ludwig, President, Radio Receptor Co., Inc.,
251 W. 19th St., New York 11, N. Y.

Asker, Gunnar, Lt. Commander, Swedish Navy,
Legation of Sweden, Washington 8, D. C.

Carpenter, Floyd R., Head, Marine Development Div., Aluminum
Company of America, 605 Southern Building,
Washington, D. C.

Cohn, Hugo, Vice President, Radio Receptor Co., Inc.,
251 West 19th St., New York 11, N. Y.

Corrigan, W. E., Vice President, American Locomotive Co.,
30 Church St., New York, N. Y.

Edgar, Herbert C., Asst. Manager, Govt. Sales, Radio Corp. of
America, Camden, N. J.

Elliott, Alec., Federal & Marine Division, General Electric Co.
1 River Road, Schenectady, N. Y.

Fixman, C. M., Marine Engineer, Code 256 M, San Francisco
Naval Shipyard, San Francisco, Calif.

Flint, Austin C., Electrical Engineer, Bureau of Ships, Navy Dept.
Mail, 5706 Eastpine Drive (Eastpines) Riverdale, Md.

Gibbs, Everett D., Chief Engineer, Radio Receptor Co., Inc.,
251 West 19th St., New York 11, N. Y.

Hinton, Henry W., Electrical Engineer, Navy Dept.,
Mail 1118 Mace Ave., Bronx 68, New York, N. Y.

Hume, Lawrence B., Sales Engineer, Cooper-Bessemer Corp.,
440 Investment Building, Washington, D. C.

Isaacs, James, Salesman, Carswell Marine Associates,
15 Park Row, New York, N. Y.

Johnson, Roscoe R., Engineering Draftsman, San Francisco
Naval Shipyard,
Mail, 1427 Castillo Ave., Burlingame, Calif.

Linford, J. W., Sales Manager, Diesel Engine Division,
American Locomotive Co., Auburn, N. Y.

Mantell, Murray Irwin, N. A., U. S. Naval DD, Terminal Island, Calif.,
Mail, 5839 Hanbury St., Long Beach 8, Calif.

Miller, Carlyl F., Sales Manager, Industrial Electronics Division, Westinghouse Electric Corp., 2519 Wilkens Ave., Baltimore, Md.

Rengel, Joseph C., Manager Naval Section, Marine & Aviation Sales Dept., Westinghouse Electric Corp., E. Pittsburgh, Pa.

Richardson, W. S., Asst. to Executive Engineer, Falk Corporation, Milwaukee 8, Wis.

Russell, Kenneth B., Sales Engineer, RCA Victor Division, Camden, N. J.
Mail, 120 N. Hinchman Ave., Haddonfield, N. J.

Schlafman, C. J. Chf., Engineer, The Commonwealth Engineering Co. of Ohio.
Mail, 1771 Springfield St., Dayton, Ohio.

Smart, Samuel N., Manager Ordnance, Westinghouse Electric Corp., 1625 K St., N.W., Washington, D. C.

Swenson, Karl E., Mechanical Engineer, Bureau of Ships, Navy Dept.
Mail, Apt. 202, 2310 40th Pl., N.W., Washington, D. C.

Waldron, Norbert H., Administrative Asst., General Electric Co., 806 15th St., N.W., Washington 5, D. C.

Watson, A. Eugene, General Electric Co., 806 15th St., Washington, D. C.

TRANSFERRED FROM ASSOCIATE TO CIVIL MEMBERSHIP.

Dennison, Boyd C., Engineer, Bureau of Ships, Navy Dept., Route No. 3, Box 34, Vienna, Va.

Else, Ralph F., Asst. Inspector of Naval Material, American Locomotive Co., Auburn, N. Y.

Gibson, M. D., Engineer, Washington Engineering Co., 601 13th St., Washington, D. C.

Hope, Herbert Arthur, Principal Marine Engineer, Office of Supervisor of Shipbuilding, U. S. N., Quincy, Mass.

Hughes, Tom E., Gen. Sales Manager, Cleveland Diesel Engine Div., G. M. Corp., 504 Hill Building, Washington, D. C.